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TRANSACTIONS

of

American Society of Heating and Ventilating Engineers

No. 1223

FORTY-NINTH ANNUAL MEETING, 1943
Cincinnati, Ohio

THE 49th Annual Meeting of the American Society of Heating and Ventilating Engineers concluded its two-day session with a registration of 380, with a variety of subjects of vital interest to engineers presented at its technical sessions. Members registered from 29 states and three provinces of Canada.

The first technical session was opened with First Vice-Pres. M. F. Blankin presiding in the absence of Pres. E. O. Eastwood, Seattle, Wash. Albert Buenger, General Chairman of the Committee on Arrangements of the Cincinnati Chapter, welcomed the members and guests to Cincinnati, and introduced Col. Charles O. Sherrill, City Manager of Cincinnati, who also extended a word of welcome on behalf of the city government and the City of Cincinnati. Colonel Sherrill spoke in detail on the engineers' work, stating that they are a rather peculiar breed of people. He paid tribute to the engineers and contrasted them with those engaged in other lines of effort. He pointed out that engineers have the happy faculty of taking a lot of facts scattered around, picking them up, sorting them, working over them, getting the meat out of them, and then coming to a conclusion based on those facts. He then referred to a celebrated engineer with the TVA, who delivered an address in Cincinnati recently, in which he pointed out the importance of arriving at a sound conclusion as to what the basic facts are.

In conclusion he complimented the Society in holding the meeting at this time, and expressed his opinion that it was the duty of national organizations to carry on during wartime, and keep abreast with developments, so that they will be prepared not only for what takes place now but what will take place in the not too distant future, when this war is won by the Allies.

Mr. Blankin responded briefly to Colonel Sherrill's remarks, and then expressed regret that Professor Eastwood could not be at the meeting; on advice from his physician he was forced to remain in Seattle. Mr. Blankin then read the President's report.

President's Report

There has been no year in history comparable to the year 1942, either nationally or internationally, and as long as the world conflict continues perhaps each succeeding year at its conclusion may be similarly designated as affecting our liberty and our civilization. The influence of war has already been felt in engineering and in engineering societies. Practically every activity of war, whether manufacturing, transportation, combat, construction, or destruction, is of an engineering nature.

This influence is reflected in the activities of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Our members are serving in government work and in the Army and Navy. Our research activities have been influenced along lines of service in war. These activities have kept pace with growth of research work throughout the world. We have always recognized the importance of dissemination of knowledge as a function of the Society, but we have largely confined this dissemination to ourselves and the engineering profession. In times like the present we realize that our responsibilities should include the general public. Comfort, efficiency, economy, and health, under rationing conditions, are of great importance to the public.

The influence of the war is expected to be perpetual and will be felt socially, politically, philosophically, and perhaps religiously. Never in history has science ever played so important a part as in the present war, and in the future science should play a much more important part in the educational system. For generations the stamp of an educated man has been impracticability, and in our universities this idea persists in the classical divisions. Scientists have no quarrel with the man who revels in poetry, literature, music, and art. Indeed, they encourage these interests, and the scientist frequently leaves his footprints when he has made excursions into these fields.

During the past year, we have followed the usual program with due consideration for the prevailing conditions. Our Chairman of the Committee on Research will report covering the work of the Research Laboratory and cooperative institutional research. There are other Standing and Special Committees at work, and, although reports may not be submitted formally, the work of these committees advances the prestige and usefulness of the Society. The accomplishments of this Society do not reflect the deeds of one man. The Society lives and goes forward because of the self-sacrificing work of its individual members. National demands and priorities account for change in occupation and, in many instances, for dissolution of businesses of some of our members.

Membership in any society is an important consideration. I acknowledge the valuable service contributed in that connection by W. A. Russell. Similarly, I recognize the valuable services of J. F. Collins, Jr., Chairman of the Committee on Chapter Relations; of A. J. Offner, Chairman of the Guide Publication Committee; of E. N. McDonnell, Finance; F. C. McIntosh, Research; A. P. Kratz, Meetings; W. L. Fleisher, Executive; and S. H. Downs, Standards; and to the personnel of all these committees. I wish to pay tribute to the War Service Committee, under the direction of Dr. B. M. Woods, and to the chairmen and personnel of the many special committees, and to all the members of Council for indispensable cooperation.

As we start another fiscal year of our Society, we realize that the past furnishes only a foundation upon which to build the future, no matter what the accomplishments of the past may be. The future challenges the membership to a rededication to service. May the same spirit of good will and sacrifice continue to build a better, stronger, and more useful organization.

Respectfully submitted,

E. O. EASTWOOD, *President*

January 25, 1943

Council Report

Five meetings of the Council were held since the last Annual Meeting at Philadelphia.

At the organization meeting January 29, 1942, Pres. E. O. Eastwood announced his Committee appointments which were confirmed by the Council. A budget for 1942 was adopted calling for the expenditure of $80,850 and an estimated income of the same amount, representing a reduction of 19 per cent compared with the previous

year. Depositories for Society funds were selected and a certified public accountant to audit the Society's books was appointed.

Announcement of the appointment of A. V. Hutchinson as Secretary, John James as Technical Secretary, and F. C. Houghten as Director of the Research Laboratory was made.

To assist in the emergency created by war, the Research facilities of the Society were offered to the people of the United States through its Government.

The second meeting of the Council was also held in Philadelphia, but due to the illness of President Eastwood, M. F. Blankin, First Vice-President, conducted the meeting.

The invitation of Cincinnati Chapter was accepted for the 49th Annual Meeting.

An analysis of membership indicated a total of 3107 on the rolls and a review of the financial condition of the Society for the first quarter showed that collections were higher than anticipated.

Because of Dr. F. C. Houghten's call to service in the Navy as Lieutenant-Commander in the reserve, it was considered advisable to provide for an Acting Director and such other administrative readjustments as would meet the demands of a wartime program.

At the St. Paul Meeting of the Council, the Membership Committee was authorized to conduct a special campaign and the Finance Committee submitted an amended budget providing for an anticipated income of $87,600 and the expenditure of an equal amount.

The War Service Committee was directed to carry on a Fuel Conservation campaign and to assist government agencies that desired their cooperation.

The Society suffered the loss of three of its past presidents who had rendered distinguished service, Messrs. R. P. Bolton, J. F. Hale and J. I. Lyle, and appropriate memorials were adopted.

The November Council Meeting was held in Washington, D. C., and an announcement was made of the confirmation of the appointment of Prof. C. M. Humphreys, as Acting Director of the Research Laboratory.

The program for the 49th Annual Meeting was adopted and it was decided that while hostilities involving the United States continue, it would be the established policy of the Society that its meetings would be governed by the amount of pertinent technical information available for presentation, its value in war work and the time necessary to conduct the business affairs of the Society.

Amendments to the Constitution, By-Laws and Rules were approved by the Council and the rate for Initiation Fees and Dues for 1943 was adopted. Approval was also given to the expenditure of minimum round trip railroad and lower berth Pullman fare for Council Members and Chapter Delegates attending the Annual Meeting of the Society in Cincinnati.

Through the Chapter Relations Committee, speakers were supplied to Chapters requesting this service and the Committee was authorized to conduct a special study of the effect of war conditions on Chapter operations during the year.

The Council granted life membership to 16, reinstated 18 members, cancelled the membership of 253 and accepted 88 resignations. It was with regret that the loss of 23 members by death was recorded.

During the year, TRANSACTIONS Volumes 46 and 47 were distributed. The Council carried out all duties required by the Constitution and By-Laws and believes that the present condition of the Society as reflected in the reports of the Officers and several committes will clearly establish that the present condition of the Society is sound.

Respectfully submitted,

THE COUNCIL.

Secretary's Report

In handling the administrative work of the Society many unexpected and unusual problems were encountered during the past year. Employment of members was maintained at peak figures most of the year, but in some classifications there are signs of decreasing need of available manpower.

The rapid changes in location of many members engaged in war work greatly increased the work of maintaining accurate address files. Often the delay in receipt of notification of members' change of location held up delivery of Society mail and publications which is costly to the Society.

Early payment of dues was made by the majority of the membership and at the end of the year 91 per cent on the rolls were paid up and 9 per cent were in arrears, according to the report of the Certified Public Accountant. An additional 2 per cent has been collected since January 1.

Over 10,000 copies of THE GUIDE were printed and distributed, and because of the heavy demand from government agencies and others, the supply was exhausted by November. It was necessary to refuse a considerable number of orders for this edition, but this will probably be reflected in increased sales of the 1943 edition which will be ready about March 1.

Export licenses required by the Board of Economic Warfare were obtained for THE GUIDE and TRANSACTIONS after passing censorship.

An unexpected development last spring was the crisis in fuel oil distribution due to Army and Navy demands, destruction of tankers by submarines on the east coast and inadequate facilities for overland delivery by pipe line and tank car. Despite the fact that the railroads made an unparalleled record in bringing oil to the eastern area the volume was not sufficient to meet all requirements. When the decision was announced that fuel oil and kerosene would be rationed, the Society's War Service Committee inaugurated an educational campaign on fuel conservation and received the support of numerous state governors in creating *War on Fuel Waste* week. The initial effort spread rapidly over 17 states originally selected for fuel oil rationing and later received greater impetus when 30 states were listed in the rationing program. On February 1 fuel oil rationing is to be effective in the Pacific Northwest states.

A great public demand for accurate information on fuel saving methods developed and the War Service Committee responded with the publication of a pamphlet listing Ten Ways to Save Fuel. This has been widely copied and distributed throughout the United States. The Society had representatives at a series of conferences in Washington sponsored by the Fuel Rationing Division of OPA and strong criticism was offered to the Fuel Oil Rationing scheme as outlined and finally adopted. The Society representatives had no part in the development of the Fuel Rationing formula, which was a product of the OPA staff, and gave no approval to the plan.

It was possible, however, for Society members to assist rationing boards on application of the fuel rationing procedure and many compliments have been received for the work of volunteer groups notably in Illinois, Michigan and Missouri. Local Chapters have been helpful in their communities through the release of advice on fuel conservation measures through newspapers and by radio broadcast. All of this has required attendance at numerous conferences and much correspondence, as well as replies to many telephone inquiries for the services of competent engineers to survey and report on heating plants. This work of the Society has received widespread publicity and has increased its prestige as a public service organization.

TRANSACTIONS Vol. 47 was compiled and distributed during the year to 2760 members and Vol. 48 is well under way.

In the headquarters office the editorial work on THE GUIDE and 12 monthly issues of the JOURNAL was carried on under the direction of the Guide Publication Committee and the Publication Committee of the Society. The maintenance of the membership files and the correspondence involved in the election of members and the reports of Committees was more extensive than usual.

It was necessary to carry on all of these activities with a reduced staff and a word of appreciation should be given to the headquarters personnel for their fine work.

A simplification of the procedure in handling speakers bureau activities reduced the time formerly required to carry on this work and also the effective cooperation of the Chairman of the Chapter Relations Committee was most helpful.

The Technical Secretary devoted much of his time to Research Committee work which was expanded in a number of cooperative institutions.

After a pleasant association of 7 years of service with the Society, John James will take up new duties in the field of research and development on February 1, and his associates extend their best wishes for his success in a new field of endeavor.

Special mention should be made of the splendid cooperation received from the members of the many committees, who have been serving throughout the year and who have contributed so much in making 1942 unusually successful.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

E. N. McDonnell, Chicago, chairman of the Finance Committee, then presented his report.

TUSA & LABELLA

CERTIFIED PUBLIC ACCOUNTANTS
52 WILLIAM ST., NEW YORK

American Society of Heating and
Ventilating Engineers,
51 Madison Avenue,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the related Funds for the year ended December 31, 1942 and submit herewith our report.

The audit covered a verification of the assets and liabilities as of the close of business December 31, 1942 and a review of the operating accounts for the year then ended. For the period audited the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected, compared with the cash records and supported by payment vouchers; also the dues income and interest income from savings accounts and securities was accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business December 31, 1942 is submitted herewith and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks listed in the attached cash schedule and reconcilement of the balances reported to us with those reflected by the books of the Society.

The cash on hand for deposit and the petty cash were verified by count.

MARKETABLE SECURITIES

The securities, per the annexed schedule, were verified by direct communication with the Bankers Trust Company where same are deposited for safe-keeping. This asset has been included in the Balance Sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

CERTIFICATE OF INDEBTEDNESS

The certificate of indebtedness issued to the Society by Farrar & Trefts, Inc., was verified by inspection of the instrument. During the current year a payment of $36.83 on account of the principal reduced the balance due to $153.20.

ACCOUNTS RECEIVABLE

A list of the membership dues receivable as of December 31, 1942 furnished to us by the management was checked to the individual ledger cards and found in agreement with the General Ledger Control. These unpaid dues were aged and may be summarized as follows:

Dues invoiced during 1942	$4,071.38
Dues invoiced during 1941	362.50
Dues invoiced during Prior Years	309.50
Total	$4,743.38

Amounts due from Guide advertisers and other debtors were verified by trial balance of the individual ledger accounts and found in agreement with the General Ledger Control.

The reserves for dues and sundry accounts receivable found on the books of the Society are ample to cover losses that might result from uncollectible accounts.

INVENTORIES

The emblems on hand were counted by us and the inventories of paper and TRANSACTIONS were verified by communication with printers.

These inventories were priced and computed by us. The following TRANSACTIONS were reported to us.

Volume	Year	Quantity	Price		Amount
-0-	Prior Years	1350	$1.00		$1,350.00
43	1937	100	1.58		158.00
44	1938	117	1.58		184.86
45	1939	150	1.66		249.00
46	1940	134	1.25		167.50
47	1941	85	1.32		112.20
47	1941	150 (unbound)	.96		144.00
				Total	$2,365.56

PREPAID TRAVELING

There were on hand railroad scrip books having a value of $24.46 which are to be turned in for refund.

PERMANENT ASSETS

Furniture, fixtures and library are shown herein at the book values without appraisal by us; we did, however, provide for depreciation of furniture and fixtures at the rate of (10%) ten per cent per annum.

ACCOUNTS PAYABLE

Amounts due creditors were determined by inspection of the unpaid bills found on file and examination of the record of cash disbursements made subsequently to December 31, 1942.

ACCRUED ACCOUNTS

Additional compensation to employees, covering the calendar year 1942, has been accrued by us in accordance with the formula contained in the council minutes of April 20, 1941. The computation is on the basis of six (6%) per cent of the Gross Guide Income of $44,846.65.

DEFERRED INCOME

All dues prepaid both by members and candidates for membership which were determined by trial balance of the dues ledger have been included in the attached Balance Sheet as a liability under deferred income.

RESERVE FOR TRANSACTIONS

The only TRANSACTIONS remaining unpublished on December 31, 1942 was Volume 48. To cover cost of publishing this volume we have provided a reserve of $3,650.00 which is the same as the amount included in the 1942 budget.

FUNDS

An analysis of the following Funds reflecting the changes that have occurred in these accounts during the year ended December 31, 1942 is included herein:

General Fund	Endowment Fund
Reserve Fund	F. Paul Anderson Fund

There is included herein a complete financial report as prepared for the Committee on Research setting forth the financial position of the Research Laboratories as of the close of business December 31, 1942 and the results from operations of the year then ended.

Respectfully submitted,
TUSA & LABELLA,
Certified Public Accountants.

January 16, 1943

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(December 31, 1942)

ASSETS

GENERAL FUND				
CASH				
On deposit	$37,989.46			
On hand for deposit	930.68			
On hand	100.00	$39,020.14		
In closed bank		380.89	$39,401.03	
INVESTMENTS (AT COST)				
Securities (Market Value $19,685.10)		18,074.06		
Add: Accumulated interest	1,500.00			
Add: Accrued interest	3.75	1,503.75	19,577.81	
CERTIFICATE OF INDEBTEDNESS				
Farrar & Trefts, Inc.			153.20	
ACCOUNTS RECEIVABLE				
Membership dues		4,743.38		
Less: 40% for Research	1,608.84			
Less: Reserve for doubtful	1,281.94	2,890.78		
		1,852.60		
Advertisers and sundry debtors	22,580.66			
Less: Reserve for doubtful	962.66	21,618.00		
Due from Reserve Fund		46.98	23,517.58	
INVENTORIES				
TRANSACTIONS—Copies		2,365.56		
TRANSACTIONS—Paper		745.25		
Emblems		90.98	3,201.79	
PREPAID TRAVELING				
Railroad scrip for refund			24.46	
PERMANENT				
Library		300.00		
Furniture and fixtures	2,688.84			
Less: Reserve for depreciation	1,215.45	1,473.39	1,773.39	$ 87,649.26
RESERVE FUND				
Cash on deposit		4,990.02		
On hand for deposit		135.00	5,125.02	
Securities at cost (Market value $34,965.00)			34,965.00	40,090.02
ENDOWMENT FUND				
Cash on deposit			2,563.82	
Securities at cost (Market value $21,933.50)		24,413.65		
Add: Accumulated interest	400.00			
Accrued interest	55.52	455.52	24,869.17	27,432.99
F. PAUL ANDERSON FUND				
Cash on deposit				1,085.83
				$156,258.10

LIABILITIES AND FUNDS

GENERAL FUND				
LIABILITIES				
ACCOUNTS PAYABLE			$16,484.83	
ACCRUED ACCOUNTS				
Additional compensation to employees			2,690.80	
DEFERRED INCOME				
Prepaid membership dues	$14,676.56			
Less: 40% prepaid to research	5,442.40	$9,234.16		
Dues prepaid by candidates for membership		570.13	9,804.29	

LIABILITIES (*continued*)		
RESERVE FOR PUBLICATIONS		
Transactions (1942) Volume 48	3,650.00	
	$32,629.92	
GENERAL FUND	55,019.34	$87,649.26

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

RESERVE FUND		
Principal	40,043.04	
Interest earned due General Fund	46.98	40,090.02
ENDOWMENT FUND		
Principal	24,063.28	
Unexpended income	3,369.71	27,432.99
F. PAUL ANDERSON FUND		
Principal	1,000.00	
Unexpended income	85.83	1,085.83
		$156,258.10

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(*For the Year Ended December 31, 1942*)

	Actual	Budget Provision	Increases Decreases
INCOME			
Initiation fees	$ 1,755.00	$ 1,200.00	$ 555.00
Membership dues—Current	28,216.30	26,500.00	1,716.30
Membership dues—Prior	1,890.00	2,000.00	110.00
Editorial contract	15,999.96	16,000.00	.04
Profit—Emblems and frames	43.68	100.00	56.32
Profit—Reprints, books, etc.	162.44	300.00	137.56
Sales of TRANSACTIONS	677.56	1,000.00	322.44
Interest—Savings banks and securities	532.86	500.00	32.86
Guide advertising	26,004.37	23,000.00	3,004.37
Sale of GUIDES	18,349.65	17,000.00	1,349.65
TOTALS	$93,631.82	$87,600.00	$ 6,031.82
EXPENSES			
A.S.A. membership	$ 100.00	$ 100.00	$ —0—
Meetings—Annual and Semi-Annual	1,268.85	1,400.00	131.15
Chapter allowance	970.82	1,000.00	29.18
Membership promotion	348.47	250.00	98.47
Speakers' Bureau	846.06	1,500.00	653.94
Subscriptions HPAC	5,854.34	6,000.00	145.66
President's Fund	1,375.29	2,000.00	624.71
Secretary's travel	1,096.73	1,000.00	96.73
Council and Chapter Delegates' travel	3,776.95	4,500.00	723.05
Salaries	18,016.50	18,500.00	483.50
Provision for additional compensation	2,690.80	3,200.00	509.20
Rent and light	3,603.92	3,600.00	3.92
Postage	1,977.14	2,000.00	22.86
General printing	968.78	750.00	218.78
Office supplies	523.93	400.00	123.93
Addressing and address changes	154.28	200.00	45.72
Telephone	698.42	500.00	198.42
Telegraph	322.29	250.00	72.29
Professional services	600.00	600.00	—0—
Bank charges	48.83	100.00	51.17
General Office expense	583.22	500.00	83.22
Membership certificates	173.94	150.00	23.94
Provision for depreciation of furniture and fixtures	268.88	200.00	68.88
TRANSACTIONS—Volume 48	3,861.32	3,650.00	211.32
Initiation fees to Reserve Fund	1,755.00	1,000.00	755.00
Codes	121.74	750.00	628.26
War Service Committee	1,052.31	1,200.00	147.69
Special appropriation to research	5,000.00	5,000.00	—0—
Weather data report	173.47	165.00	8.47
TOTALS	$58,232.28	$60,465.00	$ 2,232.72

GUIDE EXPENSES			
Paper	$ 3,850.00	$ 4,000.00	$ 150.00
Printing and binding	14,544.12	14,000.00	544.12
Express and mailing	2,767.47	3,500.00	732.53
Engraving and art work	316.20	450.00	133.80
Advertising sales promotion	1,120.31	1,500.00	379.69
Copy sales promotion	2,643.13	3,000.00	356.87
Editorial and advertising salaries	5,200.00	5,200.00	—0—
Committee expenses	268.97	350.00	81.03
	$30,710.20	$32,000.00	$ 1,289.80
YEARBOOK	$ 1,505.17	$ 1,500.00	$ 5.17
	$ 1,505.17	$ 1,500.00	$ 5.17
TOTALS	$90,447.65	$93,965.00	$ 3,517.35

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
RESEARCH FUND—NEW YORK, N. Y.

(December 31, 1942)

ASSETS

RESEARCH FUND			
CASH			
ON DEPOSIT			
Treasurer's Account—Bankers Trust Co.	$5,323.95		
Secretary's Account—Chase National Bank	2,834.09		
Director's Account—Forbes National Bank	701.41		
Thrift Account—Bank for Savings	3,301.20	$12,160.65	
ON HAND FOR DEPOSIT			
Treasurer's Account—Bankers Trust Co.		4,183.90	
ON HAND			
Petty cash—Pittsburgh, Pa.		14.84	$16,359.39
ACCOUNTS RECEIVABLE			
U. S. Navy Department			10,000.00
WORK IN PROGRESS			
U. S. Navy contract			1,889.74
PERMANENT			
Laboratory equipment			1.00
TOTAL RESEARCH FUND			$28,250.13
RESEARCH ENDOWMENT FUND			
CASH ON DEPOSIT			
Bank for Savings		$ 568.33	
Bank of U. S. in liquidation		152.53	720.86
			$28,970.99

LIABILITIES AND FUNDS

RESEARCH FUND			
DEFERRED INCOME			
PROJECTS			
Summer cooling	$1,859.45		
Glass study	1,088.38	$2,947.83	
DUES—MEMBERS AND ASSOCIATES			
40% of 1943 dues prepaid in 1942		5,442.40	$ 8,390.23
ACCRUED ACCOUNTS			
Professional services			100.00
RESEARCH FUND			19,759.90
TOTAL RESEARCH LIABILITIES AND FUND			$28,250.13
RESEARCH ENDOWMENT FUND			
Principal		$ 600.00	
Unexpended income		120.86	720.86
			$28,970.99

Note A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

Note "B": The Research Funds as at December 31, 1942 had a contingent asset amounting to $1,608.84 arising from 40% of the Society's Members and Associates dues, payable upon collection by the latter.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
RESEARCH FUND
NEW YORK, N. Y.

(For the Year Ended December 31, 1942)

	Actual	Budget Provision (Revised)	Increases Decreases
INCOME			
ASHVE—Dues	$17,986.40	$16,000.00	$ 1,986.40
ASHVE—Appropriation	5,000.00	5,000.00	—0—
Interest	49.04	45.00	4.04
CONTRIBUTIONS			
General	4,716.75	4,100.00	616.75
Earmarked—Corrosion	525.00	525.00	—0—
Earmarked—Convectors	600.00	600.00	—0—
Earmarked—Radiation and Comfort	150.00	150.00	—0—
Earmarked—Basement study	1,000.00	1,000.00	—0—
Earmarked—1941 Navy contract	1,645.06	1,645.00	.06
Earmarked—1942 Navy contract	10,000.00	16,000.00	6,000.00
TOTALS	$41,672.25	$45,065.00	$ 3,392.75
EXPENSES			
COOPERATIVE INSTITUTIONS	$ 8,888.65	$11,700.00	$ 2,811.35
PITTSBURGH LABORATORY			
Radiation and comfort	513.17	735.00	221.83
Duct friction	181.78	180.00	1.78
Cooling load	232.72	235.00	2.28
Heating requirements	2,631.82	1,800.00	831.82
Basement study	810.65	1,350.00	539.35
Air conditioning industry	2,832.49	4,000.00	1,167.51
1941 Navy contract	2,436.10	2,400.00	36.10
1942 Navy contract	10,000.00	16,000.00	6,000.00
TOTALS	$28,527.38	$38,400.00	$ 9,872.62
OTHER			
1942 Navy contract administrative	4,512.00	4,000.00	512.00
UNEXPENDED INCOME OVER EXPENSES	8,632.87	2,665.00	5,967.87
TOTALS	$41,672.25	$45,065.00	$ 3,392.75

The appointment of the Resolutions Committee was then announced by Mr. Blankin, consisting of E. N. McDonnell, Chicago, Ill., Chairman, B. M. Woods, Berkeley, Calif., and W. A. Russell, Kansas City, Mo.

The first of the technical papers was on the timely subject, warship ventilating, heating and air conditioning.

Chairman Blankin then presented Second Vice-Pres. S. H. Downs, Kalamazoo, Mich., who took the chair. Mr. Downs introduced the next author, who presented a paper on some engineering problems of the new vegetable dehydration industry.

Chairman Blankin then adjourned the morning's session, announcing that the remainder of the program scheduled for the morning would be the first order of business at the afternoon session.

The afternoon session convened in the ballroom of the Hotel Gibson, with First Vice-President Blankin presiding. Mr. Blankin turned the meeting over to Second Vice-President Downs, who called for the paper on the performance of side outlets on horizontal ducts, which was the result of research sponsored by the Society at the University of Wisconsin, and it is one of the divisions of research work conducted under the auspices of the Technical Advisory Committee on Air Distribution and Air Friction.

Mr. Blankin resumed the chair and introduced Alfred J. Offner, Chairman of the Guide Publication Committee, who presented his Report.

Report of Guide Publication Committee

The 21st edition of the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1943 has been prepared under the stress of war time conditions. While recognizing the changes which have been made from standard practices, due to these war time conditions, the Guide Publication Committee nevertheless has retained in the various chapters the high standards and basic information that has characterized former editions. Departures from these standard practices and changes in engineering methods and design, brought about by war conditions, are outlined in a new supplement in this edition entitled, Emergency War Practices.

In accordance with the policies adopted for the issuance of a new GUIDE, approximately two-thirds of the forty-seven chapters have been reviewed and of these twenty were completely rewritten with minor revisions to eight others.

Chapter 48 has been added, entitled Abbreviations, Symbols, Standards, and includes a new compilation of state laws and codes covering design and installation requirements relating to the heating, ventilating or air conditioning of buildings.

The chapter on Thermodynamics of Air and Water Mixtures has been expanded to include some new information on pressure and temperature relations at various altitudes above or below sea level, as well as some additional examples dealing with the cooling and heating load. The material covered in the chapter on Heat Transmission Coefficients and Tables has been enlarged to include some new data on basement floor and basement wall coefficients, which were developed by recent A.S.H.V.E. Research Laboratory investigations.

The Heating Load chapter has been completely rewritten and a new design temperature zone map has been added. Tabular data giving heat gain from various sources are a new feature of the chapter on Cooling Load, and in addition the illustrative example at the end of the chapter has been expanded. Chapter 11 has been given a new title Estimating Fuel Consumption, and the material in this chapter has been simplified in order to clarify the subject of degree-days in calculating fuel estimates.

The chapter on Radiators and Convectors was completely rewritten and condensed so that recent experimental data could be added. Similarly, the material dealing with Hot Water Heating Systems and Piping is new, with charts, tables and a group of graded examples added to assist the engineer in designing a system of this type. Minor revisions were made to the chapter on Mechanical Warm Air Furnace Systems, with the inclusion of standards which were recently adopted by the industry.

Chapters 22 and 23, dealing with unit types of heaters, ventilators, humidifiers, air conditioners, and air coolers, and also attic fans, have received major revisions with emphasis on new developments in the design of such equipment. New data will be found in the chapter on Air Cleaning Devices covering problems encountered where lint is an important factor. All of the material in the chapters dealing with Air Distribution, Air Duct Design and Sound Control has been reviewed, rewritten and correlated to take into consideration the results of recent Society sponsored research investigations. Other chapters which have been completely revised are Motors and Motor Controls, Industrial Air Conditioning, Industrial Exhaust Systems, and Radiant Heating. Some information on the design of solar water heaters has been added to the chapter dealing with Water Supply Piping and Water Heating. In addition to the chapters specifically mentioned all others were reviewed and checked.

An important part of THE GUIDE is the Catalog Data Section. Much useful design data and information have been supplied by the various manufacturers. Equipment has been grouped in subdivisions for convenience in locating data in reference to a particular type of apparatus.

The Committee is sorry that the new GUIDE could not be ready for inspection at this meeting. Due to the pressure of other work on the part of some contributors, receipt of a number of the manuscripts was delayed, also printing difficulties were encountered. We expect that the new GUIDE will be delivered late in February.

Your Chairman wishes to take this opportunity to express his sincerest thanks to the members of the Committee, to John James, Technical Secretary, for his fine assistance and cooperation, to those men who have so unselfishly contributed of their time and knowledge, to A. V. Hutchinson, Secretary of the Society, for his guidance and help, to Charles N Winter, Advertising Manager, for his work on the Catalog

Data Section, and to the New York office staff. Any success the 1943 GUIDE may have is entirely due to these people.

Respectfully submitted,

ALFRED J. OFFNER, *Chairman*,
GUIDE PUBLICATION COMMITTEE

Report of Tellers

C. S. Koehler then presented the Report of the Tellers of Election as follows:

BALLOT FOR OFFICERS

President—M. F. Blankin	808
First Vice-President—S. H. Downs	809
Second Vice-President—C.-E. A. Winslow	806
Treasurer—E. K. Campbell	812
Members of Council (three-year term)	
J. F. Collins, Jr.	813
James Holt	810
E. N. McDonnell	814
T. H. Urdahl	812

Scattering votes for: *President*—S. H. Downs, W. L. Fleisher, A. P. Kratz, A. J. Offner, J. H. Walker, and C.-E. A. Winslow; *First Vice-President*—L. W. McCrea, E. N. McDonnell, and C.-E. A. Winslow; *Second Vice-President*—A. P. Kratz and F. C. McIntosh; *Members of Council*—L. T. Avery, G. C. Davis, R. E. Backstrom, C. W. Helstrom, Axel Marin, and J. H. Van Alsburg.

BALLOT FOR COMMITTEE ON RESEARCH

Three-year term	
H. J. Rose	812
L. P. Saunders	811
L. E. Seeley	813
A. E. Stacey, Jr.	814
C. Tasker	814

Scattering votes for: Albert Buenger, R. S. Dill, M. C. Giannini, B. F. McLouth, J. H. Van Alsburg and M. S. Wunderlich.
Total ballots received—833; legal ballots—814; invalid—19.

Board of Tellers,

CHARLES A. FULLER, *Chairman*
R. W. CUMMING
C. S. KOEHLER

Mr. Blankin then called for the paper on army fuel consumption studies of 1941-42. The next paper was on the performance characteristics of a coal-fired space heater.

Second Vice-President Downs then took the chair and the paper on operation of the research home with reduced room temperatures at night followed.

R. L. Davison, who is director of housing research for John B. Pierce Foundation, New York, was then called upon to describe briefly the research conducted in a group of apartment buildings in New York, known as the Hillside Apartments, a group of 1415 apartments. There are five different buildings, with different heating plants for each, and the approach was quite different from the approach in the Urbana study. An attempt was made to determine what temperature would be obtained with a reduction in fuel oil consumption to conform to the fuel rationing program. Throughout the test period of 28 days the heat was on 8 hours a day, compared with 16 hours a day last year. Operating the heating plant that way there was a saving of 42 per cent of the fuel. Analyzing the temperatures and oil consumption, it

was estimated that in January and February, utilizing a third less fuel than was utilized in previous years, the average temperature in the daytime would be about 60 F.

In practically all cases people in apartments at some distance from the boiler used auxiliary heat. For the duration of the test a field staff of seven people made temperature readings and obtained statements from tenants, which indicated that 70 per cent of the people questioned stated they were not warm enough and 30 per cent objected quite strenuously to the temperatures that prevailed.

Report of Constitution and By-Laws Committee

Albert Buenger, Chairman of the Committee on Constitution and By-Laws, gave the Committee's report. He stated that all Amendements to the Constitution and By-Laws and Regulations Governing the Committee on Research had been sent out in printed form to Society members December 21, 1942:

(New) Art. C-XIII—Research—*Section* 2. The members of the Committee on Research will be held harmless by the Society from any liability to third persons, resulting from any acts in their capacity as Committee Members or when engaged in the affairs or business of the Society. Any such liability caused by the willful misconduct of any Committee Member will not be assumed by the Society.

(New) Art. B-IV—Admission Fees and Dues—*Section* 11. The Council may establish rules applying to remission of dues to members in the uniformed service of the Government during such times when the United States is at war.

Art. B-XI—Funds—*Section* 6. After December thirty-first, and before the Annual Meeting of the Society in January, the accounts of the Society shall be audited by a certified public accountant, selected by the Council. The auditor's report shall be presented at the Annual Meeting of the Society by the Chairman of the Finance Committee, and published in the Society JOURNAL.

To be amended to read—*Section* 6. After October thirty-first, and before the Annual Meeting of the Society in January, the accounts of the Society shall be audited by a certified public accountant, selected by the Council. The auditor's report shall be presented at the Annual Meeting of the Society by the Chairman of the Finance Committee, and published in the Society JOURNAL.

Art. VI—Patents—*Section* 1. If any discoveries are made under Co-operative Research agreements *with universities or colleges* * which are deemed worthy of patent application by the Committee on Research, any such applications and patents shall be owned jointly by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University, with a share to be determined by the Committee on Research and the University, to be owned by the person who makes the discovery. The cost of securing the patents shall be borne by these parties as their several interests appear.

(New)—*Section* 2. If any patentable invention or discovery is made by reason of investigations under Co-operative Research agreement with the Department of Interior and the Society, the Department and the Society shall have the right to use the patent without payment of royalty or compensation.

At the time of application for such patent, it shall be dedicated to the people of the United States or assigned to the United States as represented by the Secretary of the Interior.

Mr. Buenger read Art. C-XIII *Section* 2 and said that it would be submitted to the Society for vote by letter ballot if approval was given by the majority of those present.

On motion of Mr. Buenger, seconded by Mr. Campbell, it was

VOTED: THAT Art. C-XIII *Section* 2 as presented be approved.

Mr. Buenger read Art. B-IV *Section* 11 and offered a motion for approval which was seconded by N. D. Adams. In the discussion, Mr. McDonald of Winnipeg questioned whether the amendment covered Canadian members. Mr. Tasker proposed that the amendment be reworded and offered the following substitute which was seconded by Mr. McDonald:

Art. B-IV *Section* 11. The Council may establish rules applying to remission of dues to members in the uniformed service of the Government during such times as his Government is at war.

On vote the amendment to Art. B-IV *Section* 11 was adopted.

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* Section 1 amended by addition of words in italics.

Mr. Buenger then proposed the adoption of Art. B-XI *Section* 6 which was seconded by Mr. Avery.

On vote the amendment to Art. B-XI *Section* 6 was adopted.

Mr. Buenger explained that the amendment to the Regulations Governing the Committee on Research had been requested by the Department of Interior and he proposed the adoption of Art. VI *Section* 2, which was seconded by Mr. Triggs.

On vote the amendment to Art. VI *Section* 2 was adopted.

Chairman Blankin said that two amendments to the By-Laws had been prepared by the Committee on Constitution and By-Laws in accordance with the resolution adopted at the 48th Annual Meeting of the Society in Philadelphia, January 26, 1942 and these amendments to Art. B-VIII *Section* 11 and a new *Section* 12 had been sent to members 30 days in advance of the meeting, in accordance with the requirements of the By-Laws. He called on Mr. Buenger to present the proposed amendments.

Mr. Buenger presented the following statement:

Those of us who attended the convention at Philadelphia last year witnessed one of the best indications of the virility, strength and democracy of this Society. At that meeting a very lively discussion was carried on covering the constitutional amendments concerning the method of nominating Society Officers and Council Members.

The matter was finally referred to the Constitution and By-Laws Committee for further study and recommendation; the instructions of the Convention requiring consultation with the membership.

Since that meeting your Committee has studied the present By-Laws and the Minutes of the Philadelphia Meeting. It has exchanged considerable correspondence. Several conferences between members of the Committee and Secretary Hutchinson were held.

From these negotiations came the "Report to A.S.H.V.E. Members" from the Committee under date of October 31, 1942.

Here again a lively interest was evident judging from the number and quality of replies received. A majority of these was in favor of Plan "A" which is essentially the plan submitted today.

A final meeting of the full committee with Secretary Hutchinson and Mr. Matthews, the legal advisor to the Society, was held in New York on December 4, 1942. At this meeting the final draft was made and submitted to the membership under date of December 21, 1942.

This draft includes many of the suggestions received.

Briefly, the proposed method of nominating the officers includes the following innovations:

1. The establishment of a chapter delegates committee as part of the constitutional structure of the Society whose duties include the task of selecting a nominating committee.
2. The Chairman of the Nominating Committee shall be the past president retiring from the Council. The other members may be any four members of the Society except officers or council members, or members of the chapter delegates committee.
3. Candidates for President, Vice-President or Treasurer shall have served at least two years on the Council.
4. At least six members are to be selected as candidates to fill the four vacancies on the Council.

The Committee has unanimously agreed on these amendments and feels that they represent not only a just and fair solution of the problem but also a good cross-section of the views of the entire membership.

Finally, your chairman wishes to take this opportunity to thank the Society as a whole for their valuable replies and suggestions, and the Committee Members for their excellent cooperation, sound advice, constructive criticism and able assistance in the preparation of these amendments.

Mr. Buenger moved that the Amendments to Art. B-VIII *Section* 11 and *Section* 12 be approved. The motion was seconded.

Chairman Blankin called attention to the lateness of the hour and said that there was no desire to curtail debate but requested those who spoke on the amendment to be brief as a number desired to be heard.

After the remarks of C. H. Randolph, Milwaukee, who opposed the adoption of the amendment, a motion was offered by M. F. Carlock, St. Louis, seconded by F. E. Triggs, Des Moines, that the meeting adjourn at 6:00 p. m. and that the discussion on the amendments be continued at 9:00 a. m. Tuesday, January 26, on vote the motion was adopted.

The session was called to order by First Vice-Pres. M. F. Blankin who requested C. H. Randolph to continue his comments on the Amendments to Art. B-VIII of the By-Laws.

T. D. Stafford, Grand Rapids, thought that Committee proposed was too small and that a matter as important as this should not be passed by a small majority and felt the question should have wide support as it involved the welfare and unity of the Society.

C. F. Boester, Lafayette, Ind., said that the matter of change in the method of choosing Officers and Council members had been fully discussed by the Chapter Delegates group and said that the canvas taken indicated that a considerable number of the Society members had expressed their opinions through instructions to Chapter Delegates.

J. F. Collins, Pittsburgh, Chairman of the Chapter Delegates Committee when requested for information on the Chapter Delegates' vote stated that 11 opposed the Amendments, one favored them and 14 did not vote.

E. K. Campbell, Kansas City, expressed the opinion that there was no real desire for a change in the method of selecting Officers and Council Members and he believed that it would be a mistake to take the nominations away from the Chapters.

E. M. Mittendorff, Chicago, thought that if a change is necessary the Committee appointed should be given some idea of what was wanted.

At the suggestion of Edwin Elliot, Philadelphia, Mr. Buenger was requested to repeat the statement he gave previously and offer any other comments that he desired. Mr. Buenger stated that the matter had been taken up with the Chapter Delegates Committee last January in Philadelphia, and again, informally, in June at St. Paul with the Nominating Committee, but no suggestions were received from either group. He indicated that the Committee on Constitution and By-Laws had discussed the possibilities of a Nominating Committee of seven but had received no comments objecting to a Committee of five. He said that a majority of the comments favored Plan "A" which had been outlined by the Committee, but many had favored Plan "C" which provided for a Nominating Committee entirely of past presidents. It was the Committee's idea that the Chapter Delegates could select four members of the Committee and the plan in no way removed the power from the Chapter Delegates to select the officers. It was believed that a contest for Council Members would stimulate voting as past custom indicates that less than 25 per cent of the members vote. In closing Mr. Buenger pointed out that the Committee had no system to advocate. It had been requested to turn out an amendment to the By-Laws, after consultation with the membership and study the minutes of the Chapter Delegates Meeting and the discussion at previous Society meetings on the subject. The Committe felt in presenting these amendments to Art. B-VIII, that they were expressing the wishes of the Society.

Mr. Avery requested Chairman Blankin to read the directions given to the Committee by the Society under the resolution adopted by the Society in January 1942 and this was done.

C. S. Koehler, New York, expressed the feeling that the Committee had carried out its assignment but felt that the subject should go to the membership for longer study so as not to tie up a meeting. He also stated that he had definitely indicated opposition to a five-man committee in advance of the meeting.

Chairman Blankin then put the question for adoption or rejection of the Amendments to Art. B-VIII *Sections* 11 and 12.

On vote, counting of hands showed 20 in favor and 40 opposed. The Chairman declared the motion lost.

Chairman Blankin indicated that one amendment to the regulations governing the Committee on Research had not been voted on and Mr. Buenger presented Art. VI *Section* 1 relating to patents. The addition of four words was made in connection with cooperative agreements and

On motion of Mr. Buenger, seconded by Mr. Elliot, it was VOTED to adopt the amendment.

Chairman Blankin said he wished to express the thanks of the Officers, the Council and all of the members for the fine work done by the Committee on Constitution and By-Laws and the amount of time and thought they had devoted to it and the careful consideration that they had given to all phases of the subject.

The first technical paper at the third session, which was held on Tuesday, January 26, was on the effect of convection in ceiling insulation.

The next paper was on the subject summer comfort factors as influenced by the thermal properties of building materials, which was followed by a paper on friction heads due to water flow in copper, brass and other smooth pipes.

In the absence of L. V. Teesdale, author of the paper on comparative resistance to vapor transmission of various building materials, John James, Technical Secretary of the Society, made the presentation.

The session adjourned at 12:05 p. m.

With First Vice-President Blankin presiding the fourth and final session of the 49th Annual Meeting was called to order at 2:00 p. m. and F. C. McIntosh, as Chairman of the Committee on Research, was introduced and presented his report.

Committee on Research Annual Report 1942

The work of the Committee on Research like most other enterprises, was affected by war conditions in 1942. Members of our Society were generally busier than usual, due to the national man-power shortage, and their lack of time for extraneous work was reflected in committee activities. To prevent an improper diversion of personal energy, no effort was made to conclude studies not related to the war economy.

At the Annual Meeting of the Society held in Philadelphia in January, the Council passed a resolution offering "without expense to the people of the United States through its Government, the full use of A.S.H.V.E. Research facilities, including the specially trained personnel of its Laboratory." Subsequently this resolution was transmitted to a number of government agencies and several of them expressed interest.

The Committee on Research cooperated with the Society War Service Committee in meeting with such agencies to determine the character of research work that could best be undertaken. Several trips to Washington, D. C., were made in this connection.

On August 12, the United States Navy Department renewed a previous agreement with the Society to investigate several problems concerned with the basic design of heating, ventilating and air conditioning systems for naval vessels. In order to produce results at the earliest possible date, practically all other work at the Pittsburgh Laboratory was discontinued.

Although the number of contracts with Cooperating Institutions was the greatest in the history of the Society, no great progress was made, due to lack of man power and to a proper preference for other projects currently considered more important.

For the reasons given, reports of the 23 Research Technical Advisory Committees are generally more brief than heretofore.

Navy Comments on Results from Prior Agreements

In renewing its agreement the Navy Department stated, "the results of investigations carried on by your Society under previous contracts have provided sound data which have had a marked effect in maintaining the fighting qualities of the officers and crew of naval vessels through improved design of heating, ventilating and air conditioning systems for vessels currently in operation. It is hoped that the research work currently required will be carried

out as expeditiously as is consistently possible in order that several answers required may likewise be reflected in design at the earliest possible moment."

Navy Agreement

Two phases of the Navy work have been in progress, one at the A.S.H.V.E. Research Laboratory and the other at the Case School of Applied Science in Cleveland, under a cooperative agreement. As in the past, reports of these tests will not be available to the membership but will be submitted to the Navy in confidential form.

The current project at the Case School of Applied Science is being conducted by Prof. G. L. Tuve, and is scheduled to be completed about April 1. An extension of the scope and time is under discussion.

The Pittsburgh Laboratory program required considerable exploratory work to determine procedure, and this was established late in the year. The collection of official data, about half complete at the end of the year, is expected to be ready for report late in the summer. Former Director F. C. Houghten, a Lieutenant-Commander in the Naval Reserve, was called to active duty in April, and the Laboratory work was supervised in the latter part of the year, by Prof. C. M. Humphreys of the Carnegie Institute of Technology, who was appointed Acting Director.

Technical Advisory Committees

PHYSIOLOGICAL REACTIONS—C.-E. A. Winslow, *Chairman;* Dr. Thomas Bedford, Thomas Chester, Dr. E. F. DuBois, Dr. M. B. Ferderber, E. P. Heckel, John Howatt, Dr. R. W. Keeton, C. S. Leopold, André Missenard, Dr. R. R. Sayers, Charles Sheard, C. Tasker.

The committee is a continuing group having the responsibility of presenting at various intervals summaries of current physiological researches and their bearing on engineering practice. A comprehensive report on the physiological influence of atmospheric humidity was presented at the 1942 Annual Meeting, and it has been the feeling of this committee that new material bearing on this subject was insufficient to warrant a report this year.

Cooperative studies conducted under the auspices of this committee were continued during the early part of the year at the College of Medicine of the University of Illinois in Chicago to determine the physiological reactions of individuals entering a cooled space from a hot moist environment. In October the work at this institution was discontinued to allow more time to be spent on an important government research assignment.

SENSATIONS OF COMFORT—Thomas Chester, *Chairman;* N. D. Adams, C. R. Bellamy, G. D. Fife, W. F. Friend, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber, C Tasker.

At the request of the Guide Publication Committee, Chapter 2, the Physiological Principles of the 1942 edition of The Guide was reviewed and recommendations for its improvement were submitted.

From well authenticated information it was found that with summer cooling and dehumidifying, relative humidities of as low as 17 per cent were not objectionable with dry-bulb temperatures around 80 F. Consequently, it was thought that the shaded area of the A.S.H.V.E. Comfort Chart could be extended below its present lower limit at the 30 per cent relative humidity line to 20 per cent relative humidity. However, it was decided to hold this in abeyance until such time that the Comfort Chart could be checked in other respects, with a view to the preparation of a completely amended form. This work and other recommendations for research submitted by this committee will necessarily have to wait until more important government assigned research has been completed.

A paper on the comfort with summer air conditioning contributed by this Committee was presented at the 1942 Annual Meeting.

REMOVAL ATMOSPHERIC IMPURITIES—Dr. Leonard Greenburg, *Chairman;* J. J. Burke, J. M. DallaValle, R. S. Dill, Theodore Hatch, L. R. Koller, C. A. McKeeman, F. H. Munkelt, H. C. Murphy, G. W. Penney, Dr. E. B. Phelps, F. B. Rowley, W. O. Vedder, J. H. Waggoner, R. P. Warren, W. F. Wells.

Since the previous report, this committee has been gathering and summarizing additional data and information to complete the outline of work already done in the fields of mechanical, chemical and bactericidal purification. After the completion of this outline it will be possible to indicate the gaps of knowledge requiring subsequent research. One member of the committee contributed a major revision to the chapter on Industrial Exhaust Systems for the 1943 edition of The Guide.

RADIATION AND COMFORT—J. C. Fitts, *Chairman;* A. H. Barker, L. M. K. Boelter, E. L. Broderick, R. E. Daly, J. B. Fullman, E. R. Gurney, L. N. Hunter, A. P. Kratz, C. S. Leopold, L. L. Munier, D. W. Nelson, W. J. Olvany, G. W. Penney, W. R. Rhoton, C. J. Stermer, C.-E. A. Winslow.

At the Philadelphia January 27, 1942 meeting of this Committee, it was decided that a field project should be sponsored to determine the mean radiant and dry-bulb temperatures in actual rooms heated by convection, radiator, and panel systems. Subsequently, the A.S.H.V.E. Research Laboratory developed a test procedure and technique for conducting this work and a number of observations were made. A preliminary report covering this program is now being reviewed by the Committee.

Minor revisions to equipment in the radiant test rooms at the Pittsburgh Laboratory were made during the latter part of the year with the expectation that some exploratory tests can be made. It is intended that these tests will attempt to determine how low the air temperature in a room having a considerable amount of radiant heat can be dropped and still have a normally clothed subject feel comfortable.

The cooperative work at the University of California coming under the supervision of this Committee has been continued with the view of developing experimental data which can be used to form the basis for more rationally designing a panel heating or cooling system.

INSTRUMENTS—D. W. Nelson, *Chairman;* L. M. K. Boelter, R. S. Dill, A. P. Gagge, J. A. Goff, A. E. Hershey, F. W. Reichelderfer, G. L. Tuve, C. P. Yaglou.

Recognizing the need for greater uniformity in reporting the measurement of the physical properties of a thermal environment, this Committee prepared a tentative outline of testing procedure, which was published in the form of a paper, entitled Measurement of the Physical Properties of the Thermal Environment (A.S.H.V.E. Journal Section, Heating, Piping & Air Conditioning, June, 1942, p. 382). The four basic physical properties of the thermal environment as outlined in this report were listed as relative humidity, ambient air temperature, air movement, and mean radiant temperature. This outline is primarily intended for the guidance of research laboratory workers in this field of engineering and is not to be construed as a standard with which the industry is expected to comply.

RADIATION WITH GRAVITY AIR CIRCULATION—M. K. Fahnestock, *Chairman;* R. E. Daly, R. S. Dill, A. G. Dixon, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, T. A. Novotney, W. A. Rowe.

The progress of the work of this committee in the laboratories at the University of Illinois and in other cooperating laboratories has been severely curtailed on account of the war. Practically no progress has been made during the past year to complete the parts of the program mentioned under items Nos. 2 and 3 in the 1941 Annual Report. These items covered work to be done with radiators and convectors in several cooperating laboratories and in the cold room at the University of Illinois. All laboratories have been handicapped by a shortage of personnel which could be spared for this type of work.

A study of fluid pressure losses and heat transmission rates with water at various velocities and temperatures was actively pursued by the University of Illinois during the latter part of the year. A second warm wall test booth and auxiliary equipment providing facilities for studying the performance of radiators and convectors used with hot water has been built and placed in operation. Water velocities up to 15 in. per second with 1¼ in. connections, and water temperatures up to 240 F are available. Considerable progress has been made in calibrating the apparatus and in determining fluid pressure losses. The small tube cast-iron radiator, the copper convectors and the cast-iron convector to be used in the investigation have all been received and it is expected that good progress will be made in the future.

WEATHER DESIGN CONDITIONS—T. H. Urdahl, *Chairman;* J. C. Albright, H. S. Birkett, P. D. Close, John Everetts, Jr., C. M. Humphreys, O. A. Kinzer, H. H. Koster, J. W. O'Neill, F. W. Reichelderfer.

The activities of the committee have been greatly restricted by the war which has handicapped more active participation on the part of the Chairman and several committee members A full year of record data obtained at various selected locations in the city of Washington, D. C., is now available and it is hoped that an analysis of these data will be made early next year to show temperature variations which occur within a locality. It is felt that the information obtained may have an important bearing upon the present basis of fuel rationing which is predicated to a large extent for quantitative measure upon the degree-day which takes into consideration the temperature for a locality as a whole, though it is known that variations in the total will exist depending upon the location of the weather station upon whose readings the degree-day data will be based. The data developed under direction of this committee and the U. S. Weather Bureau indicating the severity of weather conditions through combination of low temperatures and high wind velocities should also afford some analysis serving a more useful purpose as fuel rationing criteria, especially for those vicinities where high prevailing winds will increase the heating requirement far beyond that indicated by the degree-day.

HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION—W. E. Heibel, *Chairman;* C. M. Ashley, William Goodman, H. F. Hutzel, Ferdinand Jehle, S. F. Nicoll, R. H. Norris, L. P. Saunders, R. J. Tenkonohy, G. L. Tuve, D. C. Wiley.

The major objective of this Committee has been to outline a research program for accurately determining coefficients for use in connection with the testing of finned tube heat transfer surface. Items which the Committee are interested in investigating include:

1. A study of the transfer of heat through water films existing on the external surface of finned tubes of various types.
2. Determination of film coefficients for refrigerants within the tubes; (a) with pure vaporizing refrigerants; (b) with refrigerant gas having various percentages of lubricant present; and (c) with liquid refrigerant both pure and combined with lubricants.
3. Determination of the air pressure drop through coils having wetted surfaces.

COOLING LOAD IN SUMMER AIR CONDITIONING—C. M. Ashley, *Chairman;* J. H. Carter, John Everetts, Jr., F. C. Houghten, E. H. Hyde, C. S. Leopold, C. O. Mackey, R. M. Stikeleather, J. H. Walker, W. E. Zieber.

A report giving the 24-hour heat gain rates for 11 walls and 18 roofs, having wide variation in resistance to heat flow and capacity, was presented at the 1942 Annual Meeting, as a contribution of this committee. The data presented in this paper are considered important as an initial contribution towards a more accurate method of determining the cooling load of a structure. Further study of the data is necessary to determine how the observed heat flows compare with the theory relating heat flow to the orientation, heat capacity, transmittance coefficient and absorptivity of solar radiation by the outside surface of the structure.

The committee was also responsible for supervising the cooperative investigation at Columbia University covering the periodic heat flow through walls having different heat capacities and thermal resistances. A preliminary report covering this project will be available for committee review at the time of the 1943 Annual Meeting.

AIR DISTRIBUTION AND AIR FRICTION—J. H. Van Alsburg, *Chairman;* S. H. Downs, A. E. Hershey, W. W. Kennedy, A. P. Kratz, R. D. Madison, L. G. Miller, D. W. Nelson, C. H. Randolph, M. C. Stuart, Ernest Szekely, R. J. Tenkonohy, G. L. Tuve.

The activities of this Committee have been of necessity reduced, due to the shortage of adequate personnel and, therefore, the new work accomplished has not been of any great magnitude. Studies on the frictional resistance to the flow of air in round pipes at the A.S.H.V.E. Research Laboratory have been temporarily discontinued. This also applies to additional projects coming under the supervision of this Committee, which were previously assigned to Lehigh University, Case School of Applied Science, Michigan State College, University of Wisconsin, and the University of Illinois.

As a result of work conducted during the early part of this year at Lehigh University, a paper on the effect of vanes in reducing pressure loss in elbows in 7-inch square ventilating duct, was presented at the 1942 Summer Meeting. A paper,

entitled The Performance of Side Outlets on Horizontal Ducts, will be given at the 1943 Annual Meeting, covering cooperative research conducted at the University of Wisconsin.

SOUND CONTROL—J. S. Parkinson, *Chairman;* C. M. Ashley, W. W. Kennedy, A. L. Kimball, V. O. Knudsen, R. D. Madison, C. H. Randolph, A. E. Stacey, Jr., A. G. Sutcliffe, T. A. Walters, R. M. Watt, Jr.

The research program of this committee has been designed to procure fundamental data in order to establish background noise levels encountered in air conditioned spaces and to work out engineering procedures which would permit the prediction of such levels.

Cooperative work with Rensselaer Polytechnic Institute was continued this year in order to obtain additional information for predicting sound attenuation through a duct system. The paper reporting the attenuations measured in straight ducts was presented at the 1942 Annual Meeting. This paper also described test instruments and techniques for making noise measurements in duct systems. The research program at Rensselaer is being continued to investigate the attention in bends, elbows, splitters, etc.

A tentative testing procedure has been developed by the committee for measuring the amount of noise generated by air conditioning equipment. The committee is encouraging individual experimentation to check the principles outlined in this standard in order that necessary revisions to the procedure may be subsequently considered.

AIR CONDITIONING REQUIREMENTS OF GLASS—R. A. Miller, *Chairman;* C. M. Ashley, L. T. Avery, F. L. Bishop, D. A. Bridges, W. A. Danielson, H. C. Dickinson, J. D. Edwards, J. E. Frazier, E. H. Hobbie, C. L. Kribs, Jr., Axel Marin, F. W. Parkinson, W. C. Randall, L. T. Sherwood, J. T. Staples, G. B. Watkins, F. C. Weinert.

Early in the year the committee decided not to attempt any detailed research program, but to keep in touch with conditions as they arose, and if some worthwhile investigation developed, to study the possibilities of research along that line. Earmarked funds are available for additional research. After the death on July 13, 1942, of former chairman M. L. Carr, he was succeeded by R. A. Miller. The membership of the committee recorded its sentiment to the effect that its past achievements were largely due to the prominent influence and personal endeavor of Mr. Carr.

PSYCHROMETRY—J. A. Goff. *Chairman;* F. R. Bichowsky, W. H. Carrier, H. C. Dickinson, R. S. Dill, A. W. Gauger, William Goodman, A. M. Greene, Jr., L. P. Harrison, F. G. Keyes, A. P. Kratz, D. M. Little, Axel Marin, D. W. Nelson, W. M. Sawdon.

The second progress report sponsored by this committee was presented at the 1942 Annual Meeting in the form of a paper on vapor-pressure of ice from 32 to —280 F. These calculated values of vapor-pressure will be incorporated in the final formulation of the thermodynamic properties of the moist air which is one of the principal objectives of the committee. Experimental work on the measurement of the interaction constant for moist air has continued under the cooperative agreement with the Towne Scientific School, University of Pennsylvania. These measurements have been extended to cover the range 15 to 30 C. They indicate a downward revision of the tentative value 0.075 reported in the first progress report to something like 0.055 with a small negative temperature coefficient. New data will shortly be presented to the committee for criticism and evaluation with a view toward acceptance as best information available at present.

COOLING TOWERS, EVAPORATIVE CONDENSERS AND SPRAY PONDS—B. M. Woods, *Chairman;* C. F. Boester, W. W. Cockins, S. C. Coey, E. H. Kendall, S. R. Lewis, H. B. Nottage, J. F. Park, E. T. Selig, Jr., E. W. Simons, E. H. Taze.

The work on cooling towers and spray ponds has progressed along the following course since the previous report.

1. Tests on a spray tower, with several different types of nozzles (3) were finished.
2. Further work is being done, especially with the purpose of evaluating nozzle performance.

3. Test results on two types of nozzles in a spray pond have been evaluated.
4. The performance of spray towers and ponds has been presented on a unit volume conductance basis.
5. A method of measuring unit mass conductances of single elements has been proposed in collaboration with others.

FLOW OF FLUIDS THROUGH PIPES AND FITTINGS—S. R. Lewis, *Chairman;* L. A. Cherry, G. C. Davis, T. M. Dugan, Earle W. Gray, R. T. Kern, H. A. Lockhart, Axel Marin, R. F. Taylor, E. L. Weber.

The determination of proper allowances for resistances to flow of liquids of various kinds through pipes, valves, fittings, heat receivers and transmitters is the major objective of this committee. Some additional observations were contemplated at A. & M. College of Texas on the time incrustation effect of resistance to flow of water in pipes. However, there is some evidence to indicate that this incrustation is largely dependent on the kind of pipe, the quality of water, and on the number of times the system is drained.

HEAT REQUIREMENTS OF BUILDINGS—P. D. Close, *Chairman;* E. K. Campbell, J. F. Collins, Jr., E. F. Dawson, W. H. Driscoll, H. M. Hart, E. C. Lloyd, H. H. Mather, H. King McCain, M. W. MacRae, C. H. Pesterfield, F. B. Rowley, R. K. Thulman.

Consideration was given to the continuation of the basement heat loss study, the initial results of which were reported in the paper on heat loss through basement walls and floors, which was presented at the 1942 Annual Meeting. In the continuation of this study attention is being given to the effect of insulation in reducing condensation on basement walls in the summer, to the determination of ground temperatures below the basement floor and to check tests on basement floor losses in actual residences and in other locations.

At a meeting of the committee in Philadelphia on January 27, 1942, it was decided to take steps to develop an A.S.H.V.E. Short-Cut Method of Heat Loss Calculation which would be based on the current Guide procedure and which would supplement but not take the place of the more accurate Guide method. Such a chart, applicable to one, one and one-half and two story houses has been developed and appears to give results approximating those obtained by the Guide method. An average residence heat loss problem can be solved in a matter of minutes by means of this chart which is being considered by the committee.

A paper bearing on the activities of this committee on attic temperatures, ventilation and heat losses appeared in the A.S.H.V.E JOURNAL SECTION, *Heating, Piping & Air Conditioning,* November, 1942, p. 694.

FUELS—R. A. Sherman, *Chairman;* R. M. Conner, R. S. Dill, R. B. Engdahl, A. C. Fieldner, L. N. Hunter, S. Konzo, W. M. Myler, Jr., H. J. Rose, C. E. Shaffer, T. H. Smoot, R. K. Thulman, T. H. Urdahl, E. C. Webb.

This committee held its organization meeting at the 1942 Annual Meeting. A full discussion of projects in the utilization of fuels in heating and ventilating was held, and a program of action on the subjects of pulsation in oil-burner flames, of correct methods for the measurement of flue-gas temperatures in heating equipment, and of the scaling of metals in heating equipment was outlined. Because of the pressure of other activities related to the war, little progress on any of these lines was made during the year.

To cooperate with S. Konzo and A. P. Kratz, who were consultants to Major L. C. McCabe of the U. S. Engineers, a subcommittee consisting of H. J. Rose, W. M. Myler, Jr., and R. S. Dill was appointed to assist, when called upon by Mr. Konzo, in outlining methods for estimation of fuel requirements for army camps.

The Subcommittee on Papers, headed by Mr. Konzo, reviewed and submitted to the Society two papers on fuels for the 1943 Annual Meeting.

SUMMER AIR CONDITIONING FOR RESIDENCES—M. K. Fahnestock, *Chairman;* Emerson Brandt, E. D. Milener, F. G. Sedgwick.

No active research has been conducted under the auspices of this committee during the current year, and none has been suggested for the immediate future. However, a comprehensive outline of work to be done at some future date, previously prepared, may be referred to in the 1941 booklet Programs of the Research Technical Advisory Committees.

SORBENTS—F. R. Bichowsky, *Chairman;* John Everetts, Jr., Ralph Fehr, John A. Goff, W. R. Hainsworth, C. H. B. Hotchkiss, J. C. Patterson, G. L. Simpson.

This is a new committee appointed in 1942 to survey the field of air drying agents and to act as a central group for the clearing of technical information concerning adsorbent and absorbent techniques. It will be the function of this committee to gather and recommend for publication, data on the properties of various drying agents. Fairly complete thermodynamic data on the properties of lithium chloride and lithium bromide solutions are available, but similar information regarding calcium chloride, calcium bromide and other sorbents seems to be lacking. This committee is planning to meet at the time of the 1943 Annual Meeting, where it is expected that definite plans will be outlined for future activity and progress.

CORROSION—L. F. Collins, *Chairman;* H. E. Adams, N. D. Adams, J. F. Barkley, W. H. Driscoll, T. J. Finnegan, W. Z. Friend, E. W. Guernsey, W. E. Heibel, A. R. Mumford, R. R. Seeber, E. T. Selig, Jr., F. N. Speller, C. M. Sterne.

During the summer a cooperative agreement was concluded between the Society and Carnegie Institute of Technology to study the mechanism by which carbon dioxide entrained with steam becomes dissolved in the condensate formed in operating steam heating equipment. The actual work on this project started in September. Complementary studies along the same line are being conducted by The Detroit Edison Co. The latter are not yet complete. Even so, the results obtained appear to indicate two important points:

1. That in operating equipment, where the vapor space is not vented, steam and carbon dioxide do not always form a homogeneous mixture.

2. The heterogeneity of the gas mixture is conditioned by the velocity of flow of steam through the unit.

It is believed that both of these studies will, within the next year, clear up much of the confusion presently identified with the subject.

It was anticipated that during the year the committee would assemble considerable data on corrosion protection provided by suitable paints. Due to the pressure of war work on those committee members assigned to the problem, practically nothing was done. It is planned to carry this problem for another year.

AIR CONDITIONING IN INDUSTRY—W. L. Fleisher, *Chairman;* L. T. Avery, Dr. A. R. Behnke, Dr. Leonard Greenburg, W. E. Heibel, L. L. Lewis, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, C. Tasker, R. M. Watt, Jr., H. E. Ziel.

Work was conducted at the A.S.H.V.E. Research Laboratory early this year on the physiological reactions of persons to high temperature conditions. This work was partially planned to conform to the interest of the U. S. Navy and later continued in the extension of the Navy agreement.

A paper approved by this committee covering the physiological reactions applicable to workers in hot industries will be presented at the 1943 Annual Meeting.

INSULATION—E. C. Lloyd, *Chairman;* H. King McCain, J. D. Edwards, Paul McDermott, W. T. Miller, E. R. Queer, F. B. Rowley, G. L. Tuve, J. H. Waggoner.

This committee has not met during the year 1942. However, the members individually have given a great deal of thought to the problem of utilizing the new test code covering the guarded hot plate for developing better conductivity values for use of the Society.

It is true that the events of the year have to some degree made necessary a curtailment of committee activities, but it is certain that much is to be done and the committee hopes during 1943 to:

1. Study the need for research in the insulation field.

2. The newly adopted joint committee test code covers testing of insulating materials only by means of the guarded hot plate. In order to make possible the testing of all materials used as insulation and in building construction, it will be necessary to enlarge the scope of the code so as to properly test all types of products. Therefore the committee will study the need for enlargement of the scope of the joint committee test code so that it may embrace the testing of all available thermal insulating materials.

3. Establish a recommended procedure by which qualified laboratories may conduct the testing of all available thermal insulating materials under the proposed enlarged joint committee test code.

4. Consider the need for completely replacing the present Guide conductivity values with new values based on the proposed enlarged joint committee test code and on commercial materials currently available.

HEAVY DUTY AIR HEATING FURNACES—E. K. Campbell, *Chairman;* A. P. Kratz, W. J. MaGirl, B. B. Reilly.

Late in November this committee was appointed to analyze information on heavy duty furnaces with a view to determine whether sufficient data are available and whether there is a need for the Council to appoint a code committee. It is expected that this committee will have a report to submit at the time of the 1943 Annual Meeting.

Contributors to Research

Contributions in support of the research work of the Committee on Research are hereby acknowledged:

Financial Contributions

Aerofin Corp.; Air-Maze Corp.; Allegheny County Steam Heating Co.; Harry Alter Co.; Aluminum Company of America; American Air Filter Co., Inc.; American Metal Products Co.; American Pulley Co.; American Radiator & Standard Sanitary Corp.; American Rolling Mills Co.; Anemostat Corp. of America; M. K. Arenberg; Auditorium Conditioning Corp.; Automatic Burner Corp.; Barber-Colman Co.; Barnes & Jones; Belco Mfg. Co.; Bell & Gossett Co.; Bethlehem Steel Co.; Black & Decker Mfg. Co.; Blue Ridge Glass Corp.; Buffalo Forge Co.; *California Redwood Association;* E. K. Campbell Heating Co.; Carpenter & Paterson, Inc.; Carrier Corp.; Chamberlin Metal Weather Strip Co.; Chase Brass & Copper Co.; Chicago Pump Co.; Circulators & Devices Mfg. Corp.; Cole-Sullivan Engineering Co.; Crane Co.; Dallas Engineering Co., Inc.; Detroit Lubricator Co.; Detroit Stamping Co.; Dole Valve Co.; Doyle Vacuum Cleaner Co.; Dravo Corp.; C. A. Dunham Co.; Economy Pumps, Inc.; Electric Furnace Man, Inc.; Ellison Draft Gage Co. (In memory of Albert O. Ellison); Enterprise Engine & Foundry Co.; Forslund Pump & Machinery Co.; Fulton Sylphon Co.; G & O Mfg. Co.; Gar Wood Industries, Inc.; Garden City Fan Co.; Grant Wilson, Inc.; Greene, Tweed and Co.; Grinnell Co., Inc.; D. W. Haering & Co., Inc.; Hagan Corp.; *Heating, Piping & Air Conditioning Contractors National Association;* Ilg Electric Ventilating Co.; Illinois Engineering Co.; Illinois Testing Laboratories, Inc.; Ingersoll Steel and Disc Division of Borg-Warner Corp.; Inland Steel Co.; *Institute of Boiler & Radiator Mfrs.; Insulation Board Institute;* Iron Fireman Mfg. Co.; Johns-Manville Corp.; S. T. Johnson Co.; Keystone Asphalt Products Co.; Kieley and Mueller, Inc.; Laclede Steel Co.; Leslie Welding Co.; Liquidometer Corp.; R. C. Mahon Co.; Marley Co., Inc., Jas. P. Marsh Corp.; Masonite Corp.; May Oil Burner Corp.; Mellish & Murray Co.; Modern Engineering Co., Inc.; Modine Manufacturing Co.; Morrison Steel Products, Inc.; Narowetz Heating and Ventilating Co.; The Nash Engineering Co.; National Cylinder Gas Co.; *National Lumber Manufacturing Association;* National Radiator Co.; *National Warm Air Heating & Air Conditioning Association;* New York Steam Corp.; Perfex Corp.; Permutit Co.; Pipe Fabrication Institute; Pittsburgh Equitable Meter Co.; *Portland Cement Association;* A. G. Pratt; Raisler Corp.; Randall Graphite Products Corp.; Reed Unit-Fans, Inc.; Reznor Mfg. Co.; J. E. Rhoads & Sons; F. C. Russell Co.; Schmieg Sheet Metal Works; Sisalkraft Co.; Spencer Thermostat Division, Metals & Controls Corp.; Standard Stamping & Perforating Co.; Steel Heating Boiler Institute; Supreme Air Filter Co.; Surface Combustion Corp.; Tempil Corp.; Timken-Detroit Axle Co.; Torrington Mfg. Co.; Trane Co.; Union Electric Company of Missouri; United States Air Conditioning Corp.; United States Gauge Co.; United States Steel Corp. of Delaware; Universal Cooler Corp.; Universal Power Corp.; J. H. H. Voss Co.; Waterloo Register Co.; Waterman-Waterbury Co.; Webster Electric Co.; Weil-McLain Co.; Westinghouse Electric & Mfg. Co.; Williams Oil-O-Matic Heating Corp.; L. J. Wing Mfg. Co.; Wolff & Munier, Inc.; Wolverine Tube Division; Wood Conversion Co.; Wright-Austin Co.; York Ice Machinery Corp.

Periodicals

Fuel Oil and Oil Heat; American Artisan; Domestic Engineering; Heating, Piping and Air Conditioning; Heating and Ventilating; Ice and Refrigeration; and the Journal of Institution of Heating and Ventilating Engineers (London).

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Agricultural & Mechanical College of Texas: Frictional Flow of Water in Pipes and Elbows; Heating Requirements of Buildings. *University of California:* Performance of Cooling Towers; Radiant Heating and Cooling. *Carnegie Institute of Technology:* Corrosion in Steam Heating Systems. *Case School of Applied Science:* Air Distribution in Air Conditioned Spaces. *Columbia University:* Heat Flow Through Various Building Walls. *Georgia School of Technology:* Cooling of a Structure with Attic Fans. *University of Illinois, Engineering:* Radiator and Convector Studies; Residence Summer Air Conditioning; Air Distribution Outlets. *University of Illinois, Medical School:* Human Responses to Physiological Reactions. *Lehigh University:* Frictional Resistance to the Flow of Air Through Elbows. *Michigan State College:* Air Flow in Duct Transitions. *Oregon State College:* Heat Flow Through Wet Building Walls. *University of Minnesota:* Methods of Testing Air Cleaning Devices. *University of Pennsylvania:* Measuring Departures from Dalton's Law of Air-Water Vapor Mixtures. *Rensselaer Polytechnic Institute:* Sound Transmission in Ducts and Transitions. *University of Wisconsin:* Effect of Entering Air Temperature and Velocity on Distribution of Air in Enclosed Spaces.

Research Papers—1942

1. Measurement of the Physical Properties of the Thermal Environment, by D. W. Nelson, Chairman, Research Technical Advisory Committee on Instruments (June 1942, Journal).
2. Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts (Lehigh) (TRANSACTIONS, Vol. 48, 1942).
3. Overloading of Viscous Air Filters During Accelerated Tests, by Frank B. Rowley and Richard C. Jordan (Minnesota) (TRANSACTIONS, Vol. 48, 1942).
4. Performance of Side Outlets on Horizontal Ducts, by D. W. Nelson, and G. E. Smedberg (Wisconsin), see p. 58.
5. Physiological Reactions Applicable to Workers in Hot Industries, by F. C. Houghten, Carl Gutberlet and M. B. Ferderber (Research Laboratory), see p. 188.

Chairman Blankin complimented Mr. McIntosh on behalf of the Officers and Council for his excellent report and extended the thanks of the Society to the Committee on Research for the fine work that had been done during the past year, under conditions that were exceptionally trying.

Chairman Blankin then presented greetings from the *National Warm Air Heating and Air Conditioning Association* which were sent by H. S. Sharp, President; George Boeddener, Managing Director, and F. G. Sedgwick, Chairman of the Association's Research Committee.

Second Vice-President Downs assumed the Chair and called upon the author of the paper on physiological reactions applicable to workers in hot industries.

Chairman Blankin resumed the chair and introduced W. T. Jones, Boston, Past President of the Society who with the assistance of Past Presidents Fleisher and Giesecke, conducted the Installation of Officers.

President Blankin responded briefly and was presented with the gavel. He then called upon Dr. C.-E. A. Winslow, New Haven, Conn., Second Vice-President of the Society, to conduct the panel discussion on How to Keep Fit in

Cold Homes. Dr. Winslow called the roll of the panel members as follows: Dr. C. A. Mills, Medical School, University of Cincinnati, Cincinnati, Ohio—Physiologist; Major W. J. McConnell, Medical Corps, Ordnance Department, 333 N. Michigan Ave., Chicago, Ill.—Physician and Public Health Officer; Mrs. Ivah Deering, 1118 Cypress Ave., Cincinnati, Ohio—Housewife; Edgar K. Ruth, Manager, English Woods, 1990 Sutter Ave., Cincinnati, Ohio—Housing Manager; Robert Davison, Director of Research, John B. Pierce Foundation, New York City; R. A. Sherman, Battelle Memorial Institute, 505 W. King Ave., Columbus, Ohio—Bituminous Coal; Allen Johnson, Representative, Anthracite Industries, Inc., 405 Lexington Ave., New York City; A. Stanley Bull, Insulite Co., Minneapolis, Minn.—and chairman, Technical Committee, Insulation Board Institute.

Chairman Winslow opened the discussion by stating that it would be an informal round table talk in which the members of the panel will participate and at suitable intervals discussion will be open to those in the audience. First, he requested the testimony of panel members on the subject: How to Keep Fit in Cold Homes.

Mrs. Deering responded by saying that the problem in Cincinnati was somewhat different from the Eastern Seaboard and her recent investigations revealed that homes in the vicinity had mean temperatures around 70 to 71 deg, that public buildings have not lowered their temperatures and a survey indicated that 25 public buildings during the past week showed an average temperature of 79 deg.

Mr. Bull said that a temperature of 31 deg below zero was experienced in Minneapolis earlier in the week and that inside temperatures were lower where people used rationed fuel while those burning unrationed coal and gas were maintaining 70 to 71 deg.

Mr. Davidson stated a study of War Fuel Rationing was conducted at the Hillside Homes, Bronx, New York, which consists of 1,415 apartments and houses about 5,000 people. Under fuel rationing it was found that even with auxiliary heat the day time temperature averaged 66.7 F, auxiliary heat sources probably accounted for from 3 to 5 F of this temperature.

In connection with fuel rationing program there has been a great deal of talk about 65 F. In most apartment houses, and, in most single-family houses, if they burn a third less fuel they can't get the 65 deg temperature. In New York City district, apartment owners have been fined for not keeping 65 deg; the Federal Government, on the other hand, says they have to get along with a third less fuel. The belief that 65 F can be maintained with fuel rationing is a great mistake and ought to be corrected.

People in apartment houses call up the health authorities if 65 is not maintained in their apartments. The apartment owner is called into court and fined, while, if he saves a third of the fuel, he cannot in many cases possibly maintain the 65 F.

Chairman Winslow indicated that he burned 6,200 gal of oil last year and his allotment was 3,400 for this year. His thermostat setting was 58 in the daytime and 52 at night and it was possible to keep one room comfortable by using a coal grate.

Major McConnell said that temperature had been greatly over-emphasized and that people who expected to maintain a 65 deg temperature with the amount of fuel oil given to them had been unable to do so.

R. K. Thulman, Washington, D. C., thought that the discussion was getting into the fuel oil rationing method which could not be outlined in three minutes. He stated that publicity of OPA had not contained references to the 65 deg temperature as being possible under fuel oil rationing.

Chairman Winslow explained that there was no intent to give the impression that the Government had guaranteed a 65 deg temperature under fuel rationing, but statements made by its various spokesmen on fuel oil rationing had given the general impression that 65 deg was what people could expect.

E. K. Campbell, Kansas City, Mo., made the statement that the 65 deg temperature is made official in the auxiliary rationing form issued by OPA.

Chairman Winslow said that the next question that suggests itself was the prospects of relief in the shape of additional fuel oil and the answer indicated is no, so that consideration should be given to the alternatives that are available in connection with the coal supply in regions that are now short of oil.

Mr. Johnson said that it is possible for the anthracite industry to supply sufficient coal to meet all demands. The industry, however, cannot plan ahead because there is no settled policy with respect to conversions. The response to the campaign to buy coal last summer was very successful. The essential need is to permit the industry to adopt an orderly plan of production rather than work on a hand-to-mouth policy.

Mr. Sherman indicated that the bituminous coal situation was comparable to conditions in the anthracite field. The campaign last summer to have both domestic and industrial users store coal was very effective and the bituminous industry made an all time production record. The New England area seemed to be the critical one and transportation is the key to the situation.

Mr. Johnson expressed the opinion that the 1943-44 heating season will be the most critical and that definite plans and policies should be made now to enable the fuel industry to foresee what is required, as production could not be stepped up at a minute's notice. He expressed the opinion that unless compulsory conversion was required no one would do anything after March 15, particularly if they had struggled through the winter with supplementary heat, shut-off rooms and other privations.

Mr. Thulman said the Government was being criticized for not requiring householders having grates to convert, or under certain other specified conditions, when as a matter of fact the Government has been able to find no legal way for carrying out such a program.

It was Mr. Johnson's opinion that there are 300,000 or more homes with available grates and some way should be found to put them into service.

Mr. Thulman explained that a householder has the inalienable right to freeze himself to death if he wants to and while the Government could legally restrict or limit the amount of fuel oil he can purchase, there is no way to force conversion to another fuel whether or not he has grates in the basement.

Mr. Sherman asked the question whether all commercial or industrial users of fuel oil have been compelled to convert, so that the equivalent amount of oil could be given to homes.

Chairman Winslow thought it would be possible for the Government to announce as a policy that fuel oil rations would not be provided for persons who could not prove that it was impossible or too expensive for them to convert to another fuel.

Mr. Johnson's belief was that if fuel oil could be denied to commercial users it could be denied to domestic users, and he also thought that a home owner should be required to choose whether he wants petroleum products for his automobile or his oil furnace.

Chairman Winslow thought it was too late now to do very much in the way of conversion of heating equipment, but said that if the speakers are right, it can be assumed that this problem will be even more serious next year and he felt that anything that can be done legally to stimulate conversion should be done.

J. C. Miles, Cleveland, Ohio, offered the suggestion that anthracite fuel should be sold in a form that would make its use more economical and efficient. He considered great progress would be made if the larger sizes of coal were eliminated.

Mr. Harvey said that there are millions of tons of very good bituminous coal that could be used in fireplaces and he thought that it would be advisable for people to use some of it rather than cannel coal, which not only has a high ash content, but is also one of the smokiest.

Mrs. Deering asked if it is true that the offices in public buildings and apartment houses in the middle west are still overheated, while those on the east and west coasts need heat. She wondered whether the amount of coal could be limited in the middle west if people are using it extravagantly.

Mr. Sherman indicated that the matter of supplying coal in the middle west was not a serious problem, but it might be desirable to reduce temperatures and avoid overheating.

Mr. Bull said that most of the discussion had centered around coal and oil, and pointed out that in Minneapolis, a large percentage of the homes are gas heated, while in the rural areas wood is used extensively.

Chairman Winslow said that the discussion indicated that a substantial proportion of homes on the east and west coasts will be cold for the next six or eight weeks, and many will probably be cold next winter. The question now arises, what can be done about it?

Mr. Bull then presented figures from the University of Minnesota and the U. S. Bureau of Standards, which showed possible savings in the annual fuel bill ranging from 10 to 60 per cent by the use of weather-stripping, double windows, and by insulation of various thicknesses. He said that one of the main bottlenecks was the shortage of skilled labor, and another factor was the time required to make any structural changes. In an emergency he said the application of insulating material over single glazed windows had been used and he indicated that the saving, in the temperature zone of Chicago, was approximately 1 gal of fuel oil per square foot of glass surface.

Mr. Davison thought that such application on south windows would hardly show the saving indicated.

Another translucent material often used in greenhouses was mentioned by Mr. Johnson for application over windows.

Mr. Thulman said that he had used solid panels over his windows and he indicated that the cost of covering 150 sq ft of window surface and door area was less than $10.00.

Mr. Sherman thought that the use of solid panels would require greater use of lights, although we are urged to conserve electric power.

Mr. Downs thought that savings as high as 60 per cent were too optimistic.

Wharton Clay, New York, N. Y., brought out the point that mineral wool had solved the problem for many houses already built, and he said that of the 2,400,000 houses in 33 states heated with oil, over one million have no insulation, according to Government estimates.

W. L. Fleisher, New York, N. Y., considered that the greatest public interest in the subject of heat reduction was from the health standpoint and he thought that people would have to adjust themselves to lower temperatures.

Chairman Winslow then said that the term *cold* must first be defined.

Dr. Mills offered the thought that a person is cold when he begins to shiver. He explained that when a person feels cold it is time to put on more clothes, or poke up the fire. He believed that more clothing was the easiest means of providing relief to the American people in the fuel crisis. He thought that health would be improved if people were to adapt themselves to lower indoor temperatures. The British principle of providing lower indoor temperatures, lessened the contrast between inside and outside, and they wear almost as much clothing indoors as they do out doors. The American people have gone in another direction and have built up indoor temperatures to 80 deg, so that they take off clothing indoors. The result is that respiratory ailments are common. He warned against chilling, because it promotes respiratory infection, and he said that living in cold houses should require more clothing, particularly of the extremities.

Major McConnell said that before reaching the point of shivering there are several general principles which can be used to improve room conditions.

Mrs. Deering thought that Dr. Mills' observation that silk stockings had no place in cold homes, would be rather hard on the women. She also pointed out that English women take a great deal more exercise than American women.

Chairman Winslow said that the human body is like a house, as thermal changes depend upon how many changes are put into the house, and how much heat gets out. He said that with a given activity there is a wide range of temperature adaptation depending upon the clothing worn. He said that experiments had shown a minimum comfort temperature of 72½ deg for a man in a light summer suit and light underwear, of approximately 2.9 lb of clothing, but a man in a heavy golf suit with heavy woolen underwear was comfortable at 57.5 when at rest.

Mr. Davison brought out some results from a recent clothing study developed at the Hillside Apartments in the Bronx, New York, which showed that in 45 families 42 women made adjustments in daytime clothing, while 25 men made adjustments during the same period. He stated that adjustments were also made at night with 35 women and 16 men using additional sleeping garments.

Mr. Ruth stated that in two projects that were almost identical, each containing 750 families, every possible precaution had been taken to save fuel and all ceilings had been fully insulated. Last summer an attempt was made to find out what standards should be used for winter operation and he thought that there was no Fahrenheit temperature that corresponds to a health or comfort condition. He also explained that each heating system serving three buildings supplied an average of 30 families, composed of young and old. He also stated that an educational campaign had been conducted among the tenants and he found that they were giving a high degree of cooperation.

Chairman Winslow pointed out that one method of adaptation to meet special conditions was local zone radiant heating.

C. A. Dunham, Chicago, Ill., said that from experience in this work it was found that one of the most important factors that should be taken into consideration is the per cent of humidity in each locality. It has a direct bearing on the amount of heat necessary for comfort. In London, where the humidity is high, it is perfectly comfortable in homes where temperatures are held at 65 deg. In California and Washington, D. C., a person feels comfortable at the same temperatures. Near Chicago it is very dry in the winter months, and it takes higher temperatures to keep one comfortable.

The local problem here would be dependent largely upon the per cent of humidity, and that should be carefully considered in any one locality. Humidity has an important bearing as to the sensible temperature that is required for home comfort.

A 50 per cent Relative Humidity at 70 deg temperature inside the space heated is equal in comfort to 74 deg with 10 per cent humidity.

Sub-zero temperature outside materially reduces the inside humidity through air expansion as compared with London, California and Washington, D. C., climatic conditions.

Low wall and window temperatures will also affect the sensation of comfort because of the expanded air condition, and more rapid absorption of the heat from the room.

J. N. Hadjisky, Birmingham, Mich., told of his experience with 25 American engineers who went abroad, where the temperature in the winter was from 20 to 30 below zero from Christmas to March, and he further stated that in November the windows and storm sash were puttied inside. He stated that even though supplementary radiant heaters were used and long heavy underwear was worn, the tempratures varied from 45 to 57 deg in the drafting room, and people sitting close to the outside windows found it difficult to use drawing instruments.

Mrs. Deering wondered whether there was serious danger in sleeping with windows closed, especially if various suggestions for sealing windows were carried out.

Chairman Winslow thought some of the traditions and superstitions might be discarded, although many people found it more comfortable to sleep with some air movement in a room. He indicated that many people who now live in modern housing projects have previously been trained by visiting nurses that they must have windows open in the bedroom for an hour, a tradition that goes back to the days when people never washed and it was necessary to air out.

Mr. Ruth stated that people were now getting over the habit of opening windows, and they are becoming more careful about leaving doors open and running to see the next door neighbor.

Mr. Clay inquired about the effect of wearing heavier clothing with conditions of varying temperature.

Chairman Winslow stated it was a serious problem and there has been much discussion of the subject of temperatures below the ideal optimum, but as Dr. Mills and Major McConnell had pointed out, the mass of evidence shows that people's health, comfort and efficiency are damaged by overheating.

C. H. Randolph, Milwaukee, Wis., indicated that his personal experience showed an improvement of health as a result of sleeping in an unheated room and he related an experience in connection with fuel oil rationing which indicated the large consumption of oil in some of the old homes, and a small consumption in homes where space heaters are used.

Dr. Mills brought out the point that it was not necessary to keep temperatures up in the 70's to keep it comfortable for the child under three or for the older person, even though they are producing less heat than an older child or a younger

person, but the main thing in this emergency period, when fuel is scarce, is to wear adequate clothing. Just how much clothing depends upon the individual based on his own heat production, amount of activity, etc.

Mr. Johnson in expressing a final comment felt that temperature is not very important, so far as health is concerned, because our homes not so very many years ago were fitted with fireplaces and stoves. However, insofar as comfort is concerned, and the preference of the American people, 70 deg has been chosen as the most desirable for comfort.

Mrs. Deering further commented on the open window proposition, stating that even a one inch opening would help, but she did not believe in closing the windows entirely.

Chairman Winslow in concluding the discussion pointed out that certain things were brought up and very clearly stated, that the oil burning family is likely to be in a bad way, that nothing much is likely to happen about conversion and insulation during the summer, unless the Government gets stricter with its rules and regulations concerning this problem. He then reiterated that matters of clothing and of adaptation are of immense importance. He also felt that whatever could be done to emphasize the importance of adequate clothing and the importance of adaptation to somewhat lower temperatures than have been customary will be of great advantage. He also felt that the American people will find that they will be much better off during this period with homes at lower temperatures than those they have been accustomed to. He felt that the overheating of the American homes has been a rather serious factor, and also felt that some concrete benefits might come out of this present emergency.

Report of Resolutions Committee

Be It Resolved, that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS appreciates to the fullest the inspirational leadership of Pres. E. O. Eastwood, who unfortunately cannot be present today. His stirring annual report was read with keen interest, and we as members of this Society accept with enthusiasm his challenge to a rededication to the service of building a stronger and more useful organization, which has reached the heights under his leadership during the eventful year of 1942.

Whereas John James, who has so faithfully served the Society for the past seven years, has moved on to greater opportunity and responsibilities,

Be It Resolved, that we wish him the greatest success and happiness in his new work.

Whereas the Society as an organization and through its members as individuals desires to make its services freely available to the nation in this war period, and

Acting on this principle, the President has appointed a War Service Committee to represent the Society in making this assistance available, and

Whereas the activities of this committee have included the encouragement of war-related research, collaboration with Government agencies, in the study of fuel rationing, sponsorship of conferences aimed to assist intelligent administration of the rationing program, publication and distribution of special suggestions for fuel conservation in the home, and encouragement of local chapters and members to assist the war effort wherever possible, and

Whereas it is believed that these efforts have been generally successful and that they represent an appropriate attitude on the part of the Society,

Therefore, Be It Resolved, that this action of the Society be endorsed, and that the President and the Council be urged to continue effective war service to the nation by the activities outlined and by similar activities which may be indicated as desirable in the changing conditions of wartime.

Be It Resolved, that the Society expresses its appreciation to Fabian C. McIntosh, for the outstanding job accomplished as Chairman of the Committee on Research, and be it further resolved that as this work, which is of the utmost importance to the success of the Society, has been ably supported by an outstanding group of men, who literally anticipated President Eastwood's challenge to a rededication to the service of building a better, stronger, and more useful organization, we heartily commend their efforts.

Whereas the 1943 Edition of THE GUIDE speaks for itself, and is an everliving symbol of the tremendous task performed by its able Chairman, A. J. Offner, who, with his many associates, has produced an outstanding reference book of facts developed by the very best brains in our profession, Be It Resolved that they receive our grateful thangs for their services, and Be it further resolved that the cooperating educational institutions who have so generously participated in research projects sponsored by the Society receive our thanks for their work which is the fundamental basis of the continuing success of THE GUIDE and our Society.

Also Be It Resolved that the Committee on Admission and Advancement, Earle W. Gray, Chairman, E. P. Heckel, and T. T. Tucker, be commended for their untiring efforts and investigation of new applicants to accord them the membership grades to which they are entitled by the rules of this Society, a truly difficult but important task, which they have discharged with success.

Now Whereas, the 49th Annual Meeting of the A.S.H.V.E is to be concluded tomorrow, without any further general sessions, and

Whereas, the technical sessions, the entertainment, committee meetings, and business transacted, have been of unusual interest and exceptionally stimulating to those in attendance,

Be It Resolved, that an expression of thanks and appreciation be adopted, and copies of these resolutions be published and sent to each of the persons and agencies who have contributed toward making this 49th Annual Meeting so enjoyable:

To E. O. Eastwood, the retiring President and the National Society officers and committees who devoted so much of their time and energy to the work of the Society during the past year;

To the Officers and Members of the Cincinnati Chapter and their ladies who have welcomed everyone with a cordial hospitality ever to be remembered;

To Honorary Chairman W. C. Green, General Chairman A. C. Buenger, and Ladies Chairman Mrs. G. B. Houliston, who have been most considerate of our every wish;

To the chairmen and members of all other committees of Cincinnati Chapter who have given so generously of their time to assure the success of the program;

To the authors and their associates who prepared and presented the fine technical papers, so outstanding in their scope;

To the management and the staff of the Hotel Gibson and the Hotel Netherland Plaza for their fine cooperation and service;

To the Cincinnati newspapers for their excellent coverage of this meeting, and to the trade magazines who have cooperated so ably and so generously;

To Fred F. McMinn, President of the Engineers Club of Cincinnati, for his inspiring talk;

To John F. Collins, Jr., for his very able handling of the Chapter Delegates Conference and the work therewith, and for so generously accepting a continuation of this important assignment.

Whereas, Dr. Baldwin M. Woods, a member of the Resolutions Committee, having already left for the West, the two remaining members therefore have drawn up another resolution, as follows:

Be It Resolved, that a vote of thanks be extended to Dr. Baldwin M. Woods for his very interesting, informative, and inspiring talk at the Joint Engineers Luncheon on Monday.

Mr. Chairman, I move the adoption of the report of the Committee and the supplementary resolutions.

. . . The motion was seconded by Mr. Avery and unanimously carried . . .

Entertainment

Promptly at noon on Monday, January 25, members of the Engineers Club of Cincinnati gathered with Society Members for a Joint Luncheon in the Ballroom of Hotel Gibson. Over 300 were in attendance to hear Dr. B. M. Woods, Professor of Mechanical Engineering, University of California, Berkeley, speak on the subject of The Manpower Problem in Engineering.

Fred F. McMinn, President of the Engineers Club of Cincinnati acted as toastmaster for the occasion.

In his talk Dr. Woods covered the following phases of the man power problem in engineering:

The role of the engineer as related to science and industry; the process of invention and development; inventions that have gone to war; the number of engineers required; types of young engineers in gratest demand; the place of older engineers today; methods of meeting engineering requirements through accelerated college programs and army and navy training programs; the estimated needs for the coming year in manufacturing, construction, mining, teaching, utilities and federal, state and local government departments; the post war problems of demobilization and the effect of transition from war to peacetime requirements.

On Monday evening, the Entertainment Committee provided a dinner and ice show in the Continental Room of the Netherland Plaza Hotel. A most enjoyable evening was spent by the 324 members and ladies who attended and G. B. Houliston and his Committee received many compliments on the success of this party.

Annual Banquet

The 49th Annual Banquet was held in the Roof Garden of Hotel Gibson, on Tuesday evening and the Honorable Russell Wilson, a member of the City Council and former Mayor of Cincinnati presided as toastmaster. The honored guests at the head table were: Russell Wilson, Col. W. N. Carey, President and Mrs. M. F. Blankin, Mr. and Mrs. William Green, 1st Vice-Pres. S. H. Downs, Treas. E. K. Campbell, Brig. General and Mrs. W. A. Danielson, Mr. and Mrs. A. L. Hard, Mr. and Mrs. Albert Buenger, 2nd Vice-Pres. C.-E. A. Winslow, Mr. W. H. Driscoll and Lt.-Comdr. T. H. Urdahl.

The speaker of the evening was Colonel Carey of the Corps of Engineers, U. S. Army, and Chief Engineer for the Federal Works Agency, Washington, D. C., who represented Major General Philip B. Fleming, Administrator, Federal Works Agency.

He briefly reviewed the achievements of the past year in creating a large fighting force and the more cheerful viewpoint that is apparent now compared to a year ago. He then referred to some of the economic conditions that existed in the 1920's and 30's and indicated that encouragement and confidence must be given to business through some established policy for the post war period. He stated that the kind of planning we need will start with a national survey of needed public works. A construction program, he stated, required planning and this takes time, so if we wait until the end of the war before we draw up plans and write the specifications, there will be unemployment for many months before worthwhile public projects can be started. He felt that any plans should be flexible so that they can be adapted to changing conditions. He also stated that 17½ billions of dollars from 1933 to 1941 have been spent for work relief of various sorts, and a large part of that sum was the price paid for failure to plan in advance. He also stated that if plans were carefully made, much less than the sum being spent for war in the month of January, 1943, would be ample to make good for many years to come the promise that every man shall have a job.

He concluded by saying that, if by planinng, a war over foreign tyrants can be won, it is also possible by planning to win the war against the tyrannies of depression, want and misery at home.

W. H. Driscoll, Past President of the Society introduced Pres. M. F. Blankin, who announced that the Society had a total of 81 Life Members and after reviewing some of their professional achievements, he presented Life Membership Certificates to Dr. W. H. Carrier, Syracuse, and Professor Perry West of the University of Kentucky, Lexington. Mr. Carrier expressed his appreciation for the honor conferred and read a brief statement of thanks by Professor West.

F. C. McIntosh, Chairman of the Committee on Research, presented a resolution adopted by the Committee on Research, in recognition of the services of J. H. Walker, Detroit. Mr. McIntosh then presented Mr. Walker with a handsome plaque for his service as Technical Adviser to the Committee on Research.

The Members of the Committee on Arrangements of the Cincinnati Chapter were requested to stand and the visiting members applauded them for their hospitality and the fine program of entertainment.

Following the banquet the members and ladies enjoyed dancing to the music of Jimmy Wilber and his orchestra.

Bridge

Under the direction of Mrs. G. B. Houliston, the visiting ladies were entertained during the meeting with a Dessert Bridge on Monday afternoon at the Hotel Gibson and on Tuesday a special luncheon and style show at John Shillito & Co. store.

Past Presidents' Dinner

On Sunday evening, January 24, the Past Presidents of the Society gathered for their annual dinner at 6:30 p. m. and the following were present: Dr. W. H. Carrier and W. H. Driscoll, Syracuse, N. Y.; W. L. Fleisher, New York; Dr. F. E. Giesecke, Urbana, Ill.; E. Holt Gurney, Toronto; W. T. Jones, Boston; and J. F. McIntire, Detroit.

Chapter Delegates Conference

At 7:30 p. m. on Sunday evening, the Chapter Delegates Conference was held in Parlor H of the Hotel Gibson with John F. Collins, Jr., presiding. The following Chapter representatives answered the roll call: T. T. Tucker, and L. F. Lawrence, Jr., Atlanta; A. L. Hard, Cincinnati; L. E. Seeley, Connecticut; G. E. May, Delta; B. M. Woods, Golden Gate; E. M. Mittendorff. Illinois; F. E. Triggs, Iowa; L. T. Mart, Kansas City; Ivan McDonald, Manitoba; E. G. Carrier, Massachusetts; M. B. Shea and R. E. Olds, Michigan; W. McNamara, Minnesota; G. P. Ste. Marie, Montreal; B. G. Peterson, Nebraska; C. S. Koehler, New York; F. E. P. Klages, North Carolina; T. H. Anspacher, North Texas; P. D. Gayman, Northern Ohio; E. T. P. Ellingson, Oklahoma; H. R. Roth, Ontario; B. W. Farnes, Oregon; R. D.

Morse and E. H. Langdon, Pacific Northwest; H. H. Mather and E. Elliot, Philadelphia; E. H. Riesmeyer, Jr., Pittsburgh; C. F. Boester and M. F. Carlock, St. Louis; J. A. Walsh, South Texas; F. A. Leser, Washington, D. C.; C. H. Pesterfield, Western Michigan; H. C. Schafer, Western New York, and C. H. Randolph, Wisconsin.

49TH ANNUAL MEETING PROGRAM

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL GIBSON, CINCINNATI, OHIO

JANUARY 25-27, 1943

Sunday, January 24

10:00 A.M. Registration (Foyer, Grand Ballroom)
Meeting War Service Committee (Parlor F)
Meeting Committee on Research (Parlor E)
2:00 P.M. Council Meeting (Parlor H)
6:30 P.M. Past Presidents' Dinner (Parlor G)
7:30 P.M. Informal Entertainment (Della Robbia Room)
7:30 P.M. Chapter Delegates' Conference (Parlor H)
8:00 P.M. Technical Advisory Committee on Sorbents (Parlor E)

Monday, January 25

9:00 A.M. Registration (Foyer, Grand Ballroom)
9:30 A.M. Technical Advisory Committee on Fuels (Parlor F)
TECHNICAL SESSION (Ballroom)
Welcome to Cincinnati
Reports of Officers and Committees
President's Address
Report of Council
Report of Treasurer
Report of Secretary
Warship Ventilating, Heating and Air Conditioning, by T. H. Urdahl and W. C. Whittlesey
Some Engineering Problems of the New Vegetable Dehydration Industry, by W. B. Van Arsdel
The Performance of Side Outlets on Horizontal Ducts, by D. W. Nelson and G. E. Smedberg
Amendments to Constitution and By-Laws
Report of Guide Publication Committee, by A. J. Offner
Report of Tellers of Election

12:15 P.M. Get-together luncheon (Ballroom, Hotel Gibson) Speaker—Dr. B. M. Woods, University of California, Subject—The Man-Power Problem in Engineering; Toastmaster—Fred F. McMinn, President, Engineers' Club of Cincinnati
Ladies Dessert Bridge (Parlor I)

2:00 P.M. TECHNICAL SESSION (Ballroom)
Army Fuel Consumption Studies of 1941-42, by L. C. McCabe, S. Konzo and R. E. Biller
Performance Characteristics of a Coal-Fired Space Heater, by R. C. Cross
Operation of the Research Home with Reduced Room Temperatures at Night, by A. P. Kratz, W. S. Harris and M. K. Fahnestock

4:00 P.M. Technical Advisory Committee on Cooling Towers, Evaporative Condensers and Spray Ponds (Parlor H)

6:30 P.M. Dinner and Ice Show—Netherland Plaza Hotel

8:00 P.M. Technical Advisory Committee on Cooling Load in Summer Air Conditioning (Parlor F)

Tuesday, January 26

9:00 A.M. Registration

9:30 A.M. TECHNICAL SESSION (Ballroom)

Comparative Resistance to Vapor Transmission of Various Building Materials by L. V. Teesdale

Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials by C. O. Mackey and L. T. Wright, Jr.

The Effect of Convection in Ceiling Insulation by G. B. Wilkes and L. R. Vianey

Friction Heads Due to Water Flow in Copper, Brass and other Smooth Pipes by F. E. Giesecke

12:15 P.M. Ladies Luncheon and Style Show (John Shillito Co.)

2:00 P.M. TECHNICAL SESSION (Ballroom)

Report on Committee on Research by F. C. McIntosh

Physiological Reactions Applicable to Workers in Hot Industries by F. C. Houghten, Carl Gutberlet and M. B. Ferderber

Panel Discussion—How to Keep Fit in Cold Homes, conducted by Dr. C.-E. A. Winslow with the assistance of Dr. C. A. Mills, Major W. J. McConnell, Mrs. Ivah Deering, Edgar K. Ruth, A. Stanley Bull, R. A. Sherman, Allen Johnson and Robert Davison

4:30 P.M. Nominating Committee Meeting (Parlor G)

7:00 P.M. ANNUAL BANQUET (Roof Garden, Hotel Gibson) Address, "After The War—What?" by Col. W. N. Carey, C. of E., U. S. A., and Chief Engr., Federal Works Agency; Toastmaster: Hon. Russell Wilson

Installation of New Officers and Presentation of Past President's Emblem to E. O. Eastwood

Music by Jimmy Wilbur

Wednesday, January 27

9:00 A.M. Organization Meeting of Council (Parlor H)

COMMITTEE ON ARRANGEMENTS

Albert Buenger, *General Chairman*

W. C. Green, *Honorary Chairman* A. L. Hard, *Vice Chairman*

G. V. Sutfin, *Secretary*

Committee Chairmen

Finance .... W. H. Junker
Publicity .... K. A. Wright
Ladies .... Mrs. G. B. Houliston
Banquet .... I. B. Helburn
Entertainment .... G. B. Houliston
Reception .... H. E. Sproull
Transportation .... A. W. Edwards
Inspection Trips .... M. E. Mathewson

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WARSHIP VENTILATING, HEATING AND AIR CONDITIONING

By Thomas H. Urdahl * and W. C. Whittlesey,** Washington, D. C.

THE research facilities of the A.S.H.V.E., and the consultant services donated by many of its members have been instrumental in improving and simplifying ventilation arrangements on fighting ships of the U. S. Navy. It is believed that the Society can be of further service to the war effort by acting as a medium through which to present the problem of warship ventilation to engineers engaged in the shipbuilding program.

Design facilities, fabricating plants, and shipyards are now being used that have had little or no background in naval construction. This has greatly increased production, but considerable improvement is necessary in matters of detailed design. Air conditioning and ventilation arrangements are often fitted in such a way that there are a few factors that develop undesirable results in otherwise satisfactory installations. Some of these items can be corrected when and if the ship has yard availability, but others are such an integral part of the ship's structure that proper remedial measures are virtually impossible.

Surface and submarine war machines are vital to military might, but they must be functionally designed and built to the best and most modern scientific and engineering standards to successfully engage enemy units. Traditional folklore to the effect that warship design requires a special fundamental approach is misleading—it is essentially sound engineering practice with the emphasis placed on factors rarely encountered elsewhere.

Special Military Requirements

Prime factors in a warship are ordnance, protection, and speed. Each of these factors is compromised to some extent in favor of the others because of over-all displacement limitations, and all three must be reduced because of other necessary complementary features. Consequently, it becomes vital to keep all secondary features to an absolute minimum of space and weight. Ventilating, heating, and air conditioning arrangements are complementary features fitted only to enable personnel to effectively fight the ship, and not to provide comfort conditions as usually required in commercial work. Some types of shore installations in industrial establishments might be called similar to the naval problem, although they differ in detail.

* Lt.-Comdr., USNR, Officer-in-Charge Air Conditioning Section, Shipbuilding Div., Bureau of Ships, Navy Dept. Member of A.S.H.V.E.

** Senior Engineer, Air Conditioning Section, Shipbuilding Div., Bureau of Ships, Navy Dept. Member of A.S.H.V.E.

The opinions and assertions contained in this paper are the private ones of the writers and are not to be construed as official or reflecting the views of the Navy Department or the naval service at large.

Presented at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1943.

A successful commercial design is usually judged by the enthusiasm with which the installation is accepted, and the cost per unit of service. However, these criteria are not applicable to warship ventilation installations. A system that is eminently satisfactory to forces afloat is probably unnecessarily bulky and heavy, with the result that some ammunition or other vital equipment must be left ashore. Also, it is poor economy to simplify a ventilation system by reducing its resistance to damage, with the possibility of thus losing a ship.

There are a number of considerations or design factors that are peculiar to Naval work and result in practices that are difficult for the outside engineer to rationalize. The following items are presented as an explanation for factors that are troublesome to engineers encountering Navy Specifications.

1. *Minimum Weight:* The weight and space requirements for all equipment must be kept to the absolute minimum consistent with satisfactory operation, and arrangements must be such that they restrict damage from flooding and fire. Small ventilation ducts are necessary to satisfy these requirements because they result in less weight and they afford greater protection because the piercings through watertight and armored structure are smaller. The use of small ducts requires high velocities and fan pressures; however, this is preferable to increased weight and larger structural piercings.

2. *Watertight Integrity:* Ventilation ducts must not pierce main watertight structure below the highest possible waterline in order to provide protection against flooding in case of damage. This requirement necessitates circuitous leads which, combined with high velocities, complicate duct design. Ventilation piping must be designed to give uniform flow in the ducts to avoid noise and distribution troubles. In this connection it is important to use vanes in elbows and bends.

3. *Rugged Construction:* Warship equipment must be sufficiently rugged to withstand the shock of battle action and normal naval service. The shock caused by some types of enemy action produces accelerations in the hundreds of g's. Many ventilation systems are vital to the operation of the ship and must stand this strain to keep the ship in action. Brittle materials such as cast-iron, and too stiff or too flexible construction will cause failures. The temptation to include extra safety factors for warship equipment is great when the severe service requirements are realized; however, dependability should be provided by refinement in design and attention to detail instead of the arbitrary addition of mass, which inevitably results in pyramiding weight. Also, fighting men living and working in imminent danger of sudden death are inclined to play a bit rough. Space is congested, headroom low, and all ventilation fittings are within easy reach. Any gear that will not withstand a solid blow from an object such as a baseball bat is likely to have a short life span, and because of this condition substantial ducts of heavy gage metal are necessary.

4. *Foolproof Design:* Arrangements must be as foolproof as possible, designed to require a minimum of care, and installed to be tamperproof. The function of a warship is to fight, and the less the maintenance requirements are for complementary features the more time will be available for the principal business in hand. A large proportion of the sailors are mechanics and possess an insatiable curiosity about the why and wherefore of their environment. A traditional Navy requisite is for a man to know the purpose of every pipe, valve, and fitting at his station, and how it operates. Such individuals will soon discover that a change in a splitter setting will produce an increase in their ventilation, with the result that others are deprived of their proper share. Of course, such practices cause considerable distress, and no balancing device should be used that is easily adjustable or can be altered. For years splitter dampers or adjustable mechanical volume control devices were not permitted, and systems had to be balanced entirely by altering duct sizes or installing permanent orifice plates. However, many ship-builders found this method cumbersome and tedius, with the result that balancing was glossed over or omitted. Splitters are now acceptable, but they must be of such a design that they can be fixed in place by welding or riveting to prevent a clever sailor, with a screwdriver and a pair of pliers, from readjusting a system to his own liking.

5. *Insulation Variables:* The insulation problem on steel vessels is complicated because heat not only flows by transmissions through bulkheads and decks but is also conducted long distances through the heavy steel structures. The heat from a fire room in the hold may be responsible for high structure temperatures on the second or main decks. The thickness of insulation ashore is usually chosen to give a minimum total for interest, depreciation, and operating costs for all equipment and fittings, but the thickness of insulation on a warship is chosen for a minimum total weight of insulation and equipment to produce cooling. This policy usually results in less thickness than is considered good practice ashore. The importance of good workmanship in fitting the insulation cannot be over-emphasized as thorough covering is necessary. Air quantities for cooling are limited to a point where the resultant ambient air conditions approach critical temperatures—in other words, the designed conditions are as high as we can safely go. This is done to save weight. If the insulation is poorly fitted with a consequent increase in the heat gain over the values anticipated in the design, the resultant compartment temperatures will be too high. Designing so closely to limiting conditions may be poor practice elsewhere, but is

U.S.S. NORTH CAROLINA (OFFICIAL PHOTOGRAPH, U. S. NAVY)

absolutely necessary if our ships are to meet the enemy without already being half sunk by excess equipment. Weight limitations and rigid requirements for resistance to various kinds of damage and deterioration drastically restrict the acceptable insulation materials.

6. *Heating Arrangements:* Heating systems are designed to give acceptable performance with a minimum weight. This is usually accomplished by using fin tube steam heaters installed in the ventilation ducts. These are broken down into two sections—preheaters at the intake which raise the air temperature sufficiently to preclude condensation on unlagged ducts, and reheaters to raise the temperature of the supply air to the level required for heating a large space or group of small spaces of about the same exposure. This method does not provide the individual control which many desire, but it conserves weight. Care must be taken not to provide too much heat, for if shipboard compartments are maintained at ordinary comfort levels the men will not dress properly for topside duty.

7. *Exhaust Provisions:* There is a tendency to omit positive exhaust arrangements. Natural leakage through windows, doors or skylights on shore installations may provide adequate exhaust, but most shipboard spaces are watertight, gas-tight, or at least weathertight, and access doors or hatches must be kept closed at times when ventilation is most needed. Positive exhaust is a prime essential.

8. *Weather Openings:* Weather ends of both supply and exhaust systems must be designed and located to expel sea water in rough weather and to prevent air recirculation between supply and exhaust openings. Ventilation cannot be secured in spaces such as engine rooms regardless of the weather, and sea water entering through the ducts will cause casualties to electrical equipment. Also, closing down the ventilation to living spaces during foul weather is very disagreeable and not at all conducive to a

fighting attitude. The satisfactory location of openings is difficult since warships are designed with as little superstructure as possible to reduce the area outside of primary protection, and to provide the maximum possible train for the battery. Also, director foundations and other fittings produce eddy currents that vary with the relative wind direction and force. This crowded condition and variation of turbulence often result in recirculation problems. The best way to avoid the trouble is to have supply openings on one side of the vessel and exhaust openings on the other; however, this is seldom practicable, and the designer must often make the best arrangement possible under difficult limitations.

9. *Universal Weather Design Conditions:* A fighting ship must be capable of operating in any navigable sea in the world at any season of the year. This requires cooling and heating systems that will provide adequate conditions in the Arctic in winter or the Tropics in summer, and a wider range of operating conditions is necessary than is usually encountered ashore.

10. *Corrosion Problem:* Continuous exposure to sea water and salt air result in a difficult corrosion problem, and many materials and coatings that have provided satisfactory protection ashore have failed afloat. Consequently, the Navy is skeptical of new developments and requires rigid and extensive tests of new proposals regardless of their previous record.

11. *Personnel:* The problems are somewhat simplified because all personnel are healthy male specimens, better able to adjust themselves to relatively extreme conditions than the average person. However, many individuals seem to be totally unaware of their remarkable ability in this matter and complain bitterly about the conditions to which they are subjected. This situation is at least partially chargeable to the comfort cooling standards developed by the industry for theaters, hotels and other public buildings. Considerable diplomacy is necessary in handling these complaints—it only antagonizes people to quote experiments proving that human protoplasm will remain in good working order at conditions outside of the comfort zone.

Outline of Treatment

The treatment required for the various shipboard spaces is determined by the basic use of the compartment and the type of ship on which the installation is to be made. The treatment may be roughly divided on the basis of the spaces served: (1) berthing and messing spaces; (2) working spaces; (3) control spaces; (4) heat producing spaces; and (5) stowage spaces.

The atmospheric conditions that prevail in living compartments aboard a warship have an important bearing on the health and morale of the crew, and the maintenance of satisfactory berthing and messing space temperatures is a naval necessity. Shipboard tests have demonstrated that personnel can work effectively and efficiently in extreme atmospheric conditions over extended periods of wartime cruising so long as there is a reasonably comfortable place in which to relax when off watch. A sailor's duties in war are varied, and often his free time is broken up by extra duty or alerts. Consequently, it is doubly important that living space conditions are conducive to adequate rest.

Living spaces are now cooled by means of ventilation, *i.e.*, fresh air is circulated through a vessel to provide cooling in the same way jacket water cools an engine or air cools an automobile radiator. The quantity of air provided is based on maintaining a specified temperature rise over outside air or providing a given minimum quantity per man, whichever is the greater. The exposure of living spaces and the heat producing machinery located therein varies greatly for different locations and for different types of vessels; consequently the design cannot be governed by an air change method.

The distribution problem is complicated by the density of occupation in berthing spaces, the large air quantities, and the headroom which is barely sufficient to provide clearance for personnel. Overhead distribution through ducts and registers in such spaces has resulted in objectionable drafts, and a method of diffusing air against the deck has been developed that is quite successful. Distribution for messing spaces is usually through overhead ducts fitted with diffusing terminals. The exhaust is centrally located, care being taken not to short circuit supply air.

COOLING WITH AIR

The ventilation method of cooling results in inboard conditions of about 10 F above topside air. Weather air in the tropics may be as high as 90 F dry-bulb and 80 F wet-bulb for weeks on end, and under such conditions it is impossible to maintain satisfactory temperatures in living spaces with ventilation only. A rise of 10 F dry-bulb and 4 F wet-bulb over the above conditions would result in 89 deg ET. A person exposed to such conditions will perspire profusely and sleep lightly. Although this will not produce any measurable deleterious effect over short periods, extended exposure will dull alertness, dexterity, and both physical and mental processes. One of the most striking examples of reduced efficiency is shown by a comparison of the time it takes a sailor to awake and take such action as the particular situation demands under different temperature conditions. In cool weather this is accomplished in almost nothing flat, amid loud and sundry grumblings garnished lavishly with appropriate epithets; but under tropical conditions many minutes are required for the men to crawl slowly and listlessly out of their berths and become cognizant of the situation at hand. The reactions of these bewildered individuals provide considerable amusement for their contemporaries already on watch, but in case of a surprise enemy attack this situation is anything but humorous.

MECHANICAL COOLING LIMITATIONS

Air conditioning, or more accurately, mechanical cooling, appears to be the only practical answer; but regardless of its desirability it is not generally applied because of concern over the dependability of the equipment together with the required increase in weight. A recent comparative study showed that the use of a single unit air conditioning system for a group berthing space would result in a weight increase of 88 per cent over that required for ventilation. Although this system would provide satisfactory temperatures during tropical cruising, whereas ventilation will not, the weight increase is absolutely unacceptable. The application of air conditioning to living spaces aboard warships must await the development of equipment that can compete more favorably with the present ventilation system in dependability, weight, and space requirements.

WORKING SPACES

Shops and other similar working spaces are cooled by ventilation. The quantity of air provided is based on maintaining a specified temperature rise and in providing a minimum air change if any inflammable or toxic vapors

are likely to be present. Distribution is usually through diffusing terminals in the overhead with the exhaust located over heat sources such as a forge or bending slab. This arrangement has proved satisfactory in service.

A few vital control spaces on the larger vessels are air conditioned by self-contained units to maintain ideal working conditions, and to permit these compartments to be positively sealed from any source of fire or flooding. Dual units are provided to give dependability, but each is selected to maintain an emergency condition which is barely habitable, such as 103 db and 85 wb. This reduces the required tonnage and the necessary weight and space. The controls for these units are adjusted for comfortable temperatures instead of the designed values, and the use of both units simultaneously will provide sufficient tonnage to maintain low temperatures at any time. Inasmuch as the selection is based on the battle load, one unit is usually capable of maintaining comfortable conditions whenever the crew is not at battle stations; and should one unit fail during battle, the other will maintain the emergency condition.

Hot Spaces

The concentration of heat-producing equipment in machinery spaces such as the engine and fire rooms is so great that satisfactory conditions cannot be maintained throughout every part of the compartment. Even if this were possible, its desirability is questioned, for a tremendous quantity of energy is removed by the exhaust air in the form of heat and increasing the ventilation over present standards would mean increasing this heat loss and thus reducing the plant efficiency. This point may be emphasized by equating the horsepower equivalent of the heat energy removed by the ventilation air to the horsepower required to propel the ship. On one ship this power was equivalent to that required to drive the ship at nine knots. Attempts have been made to use air conditioned control booths or cubicles for personnel, but these have met with disfavor, for it is believed that operating personnel should be constantly in close contact with the machinery. Preliminary experiments have also been conducted in the use of ventilated suits, but this meets with the same basic objection of restricting freedom of movement. The ships' engineering officers are of the opinion that the men are there to service the equipment—and are naturally opposed to radiant heat shields or any other gadget that interferes in the slightest degree with machinery accessibility or maintenance.

Spot-cooling of operating stations and normal working areas is the method used to ventilate heat-producing spaces. This is accomplished by means of a large jet or blast of supply air delivered directly to the watch stations, thereby giving the watchstanders the benefit of the weather air before it has been heated by the equipment. The air blast arrangement also reduces the effective temperature of a given wet- and dry-bulb combination by increasing the air velocity over the body. Adjustable terminals are required for this service, and they must be installed in such a way that the blast will sweep the working area and can be directed against the front of the operator's torso. If it is directed on his head or the back of his neck, discomfort will result in even the hottest compartments. Personnel served by spot-cooling can leave the *oasis* for a sufficient time to service machinery in the hot portions of the space. Portable blowers are rigged in case emergency repairs are necessary.

Exhaust arrangements in hot spaces require particular attention. The exhaust

fan capacity must be in excess of supply fan delivery not only because of the expansion of the ventilating air due to the temperature rise, but also to induce an indraft through the space access. If this is not done, hot air from machinery spaces will circulate upward through the ship and overheat the living or working spaces through which it passes. Such a condition can make a miserable ship out of what might otherwise be a well-ventilated vessel. Exhaust should be through large bell mouth terminals in the overhead above the principal heat sources.

Ventilation is vital to the operation of a machinery space, and great care must be taken to assure dependable operation. Tests have proved that many machinery spaces cannot be manned longer than a few minutes should the ventilation fail; consequently, high ambient temperature motors are used for these fans, and duplicate power circuits are often fitted to insure dependability. Also, fans should be installed in protected locations.

Considerable improvement can be made in heat producing spaces by thorough and complete lagging. Every pipe, every flange, and every valve bonnet on hot piping must be covered. Decks, bulkheads, frames, and stiffeners should also be thoroughly insulated to reduce the heat transmission and conduction through the ship's structure to cooler compartments. Every square inch of surface should be properly covered. A recent investigation showed that the omission of the lagging from a steam line flange and from about one square foot of the overhead in way of this flange resulted in the overheating of a magazine group that was located above. To repeat again, shipboard insulation is used sparingly to save weight, and it is necessary to have complete coverage. Good workmanship is absolutely necessary for effective insulation and lagging applications aboard ship. The standard procedure is to have the lagging gang cover everything that is possible to cover, and then go ahead and cover the impossible with insulating cements or any other feasible material. The practice of omitting lagging to facilitate repair should be discouraged, as it inevitably results in overheated spaces. The removal and replacement of lagging should be considered just as much a part of repairing a valve as fitting the flange gaskets.

STORAGE SPACES

Stowage spaces are treated on the merits of each case. Ventilation is used only as a last resort, as the majority of storerooms are either low in the ship or in outboard spaces provided essentially for protection, and the necessary holes for ventilation ducts constitute a flooding hazard that is always undesirable and often unacceptable. No treatment whatever is provided if there is a good chance that the material stowed will not be functionally affected. Shore standards of maintaining appearances to preserve salability are ignored. Desiccants are used where moisture removal is necessary and ventilation is used only where cooling is required. Possible explosive concentrations in compartments adjacent to inflammable liquid stowage are controlled either by ventilation or by maintaining atmospheres that will not support combustion.

Refrigeration is provided for some types of storerooms other than cold storage boxes where a definite maximum temperature must be maintained or where the boundary piercings required for ventilation are unacceptable. Gravity evaporation coils installed in the overhead are generally used for this type of

service. A manufacturer may be requested to refrigerate a space to 90 F, although the control thermostats are set for lower temperatures which can be maintained except when the ship is operating at maximum power in the tropics or in battle condition.

BALANCING IS ESSENTIAL

One of the biggest problems in warship ventilation is to get the systems properly balanced before delivery of the vessel. Balancing is the adjustment of a system to compensate for errors in the design, fabrication, and installation of the ducts to provide the proper quantity of air at each terminal. All the blood, sweat, and tears that are expended on a ventilation system are utterly wasted if this end is not achieved. There is too much of a tendency to consider an installation complete when all ducts and other fittings are in place and the fan is turning over. True, the work is over 99 per cent complete, but the system will be only partially effective, and in many cases totally unacceptable, if not properly finished by balancing. The results of unbalance are rarely apparent during the building period and preliminary trials since air quantities are based on full power wartime operation in the tropics, with all auxiliary equipment in use and a full complement living aboard. Consequently, maximum ventilation is not essential in home waters and few complaints are made until after the ship has left the building yard.

Furthermore, the possibility of more equitably distributing the available air never seems to occur to forces afloat, and in cases of unbalance additional ventilation is usually requested. Such emergency alterations, completed during restricted availability at an advanced base, invariably increase weight out of proportion to the improvement and often result in arrangements that are not properly installed to resist damage.

Adequate and accurate tests are required to properly balance a ventilation system. The testing of ventilation systems under the most ideal conditions is trying and seems to be a controversial issue; but the average system aboard a warship is believed more difficult to test than the worst system elsewhere. The complicated intakes, the large number of offsets, bends, elbows, fittings and transition sections that are necessary to run the average duct in the restricted space available all contribute to turbulence and complicate measurements. However, a simple technique has been developed for volumetric tests, based on the much maligned rotating vane anemometer, in which readings taken at various points of a system and by different individuals show remarkably consistent results.

STANDARDIZATION OF EQUIPMENT

The equipment used for ventilating, heating, and air conditioning arrangements must be simple, rugged, light, reasonably quiet, and economical as to power requirements. As the supply of critical materials becomes more acute and rationing becomes more detailed, it is necessary to plan accurately for future needs in order to obtain the required allocations. It is also vital to place large orders to permit manufacture on a quantity production basis. This can only be done by standardization and anticipating requirements. In the present building program it has been found necessary to maintain stocks of

ventilation equipment to avoid delaying completion dates on new construction, and to expedite conversions and repair.

Stocks of standardized equipment must also be maintained at advanced repair bases to care for battle damage. This requirement becomes more and more urgent as the frequency of engagements increases. The speed with which damaged ships can be reconditioned for battle will have an important effect on the course of action, for with anything approaching equal forces in a given theater, victory may be determined by the speed with which the shipyard can refit damaged vessels. This one factor is ample reason for equipment standardization and a large investment in equipment stocks.

Furthermore, standardization is a great boon to the designer. In the days of competitive bidding on individual manufacturer designed equipment, the ventilation and heating contract was rarely let until after the deadline for working plans, with the result that the designer never knew just what his equipment dimensions or performance characteristics would be. Now the designer knows exactly what he will get, what it will do, and its dimensions. Obviously, a standardized fan line will limit the range of selection, and it is now necessary to design the ducts to suit a given fan, rather than to choose a fan to suit a given duct design. This policy created some confusion at first but is now showing good results.

Ventilation fans were the most difficult to develop and standardize. The success of this venture was made possible by the cooperation of the fan manufacturers and their willingness to expend funds in the development of a fan to meet Navy requirements. There was a trying interlude when the Navy did not seem to know exactly what it wanted, or whether it would be able to purchase such fans under the competitive schedules after development; however, everyone kept plugging away on the problem and an excellent product finally emerged. This fan is small and compact, satisfies the Navy requirements as to ruggedness, operates direct connected at high speed with a low noise level, is capable of developing high pressures, but can operate satisfactorily down to free delivery, is readily adaptable to watertight applications, has limit load power characteristics, is so efficient that power requirements are very low, and is capable of being installed in much less space than was formerly required. A recent development provides a simple mechanical volume control.

All in all, this standard fan description probably sounds like the answer to a maiden's prayer. It is. Before its advent, all attempts to use high pressure fans had produced noise levels in living and control spaces that were actually detrimental to the crew's health and adversely affected ship operation. During this period the reception of ventilation representatives aboard ship may be described as *restrained*. They were assigned staterooms so noisy that conversation was impossible—in one case the noise level on the berth pillow was measured at 93 db. The use of the new fan has practically eliminated the noise problem, and personnel afloat have returned to their time-honored complaint of uncomfortable temperatures.

Other equipment is also standardized or in the process of being standardized. Standard plans for heaters, convectors, watertight closures, diffusing terminals, and duct elbows are now available, although development is continuing on this equipment. Standardization is the key to equipment procurement, and it is anticipated that difficulties will vary inversely with the degree of standardization

that can be developed and maintained during the present emergency. Manufacturers will be called upon to assist in the standardization program as it is believed that this will not only develop better equipment for the Navy, but will also ultimately result in improved equipment for the industry as a whole.

Considerable difficulty has been experienced with evaporator coils furnished for air conditioning installations. The manufacturers vary widely in their coil selections for a given application, and it has been necessary to replace some coils in order to develop the required tonnage. Evaporator coil workmanship has often been disappointing, and some types foul so quickly and are so difficult to clean that their usefulness aboard ship is seriously curtailed. The Navy has developed standard blast coils, and unit evaporators for shipboard are to avoid these difficulties. The use of these standard coils, and a selection technique that incorporates the application of effective temperature provides satisfactory installations that insure a maximum output per pound of equipment.

Conclusion

Progress in warship construction can only be made by providing more punch per pound, and a naval engineer should strive to develop lighter, simpler, and more foolproof methods to accomplish this end. Conventional practices should be continually analyzed from the functional standpoint and revised wherever commensurate savings in weight or space are possible, or where dependability can be improved.

Whenever a member of this society is solicited for technical assistance regarding Naval construction, he should consider it his patriotic duty to apply the principles outlined herein. Sailormen are fighting to the death on the high seas in our battle with the enemy. It is our responsibility to provide them with the best fighting ships it is humanly possible to build.

DISCUSSION

J. W. MARKERT, Washington, D. C. (WRITTEN): The authors of this paper have successfully overcome the problem of presenting a subject which may be considered confidential in a non-confidential manner. It has frankly presented the problem in order to encourage improvement and interest. The purpose of this discussion is to briefly compare Naval and Maritime Commission practice in a general manner. An effort will be made to parallel the contents of the paper in order that differences and similarities may be easily discernible.

Today practically every merchant vessel is a naval auxiliary and carries a certain amount of defensive armament and ordnance to protect itself against attack from the sea and air. Because of this, there is more of a tendency to follow Navy standards than usual.

The Commission has attempted to stress simple, efficient designs in order to save weight and space as well as unnecessary maintenance and repair. In most cases static pressures and air velocities are somewhat lower than those found on Naval vessels because of the limits of generator capacity, and lower ambient noise level. Space requirements are somewhat more liberal on merchant vessels, but by no means comparable with that on land. Under normal circumstances it is possible to design systems so that ducts do not pierce the main transverse water-tight bulkheads.

Regarding ruggedness of construction and corrosion, it must be said that both

Naval and Merchant ships sail the same seas, use the same air, and weather the same storms. While the merchant vessel ordinarily is not subject to the same amount of shock, present day activity has materially increased the importance of this factor. There has been a growing tendency in merchant ship design to eliminate cast iron. It is interesting to note that in several instances the substitution of sheet steel has resulted in decreased production costs and an increased production, as well as a more reliable product. The corrosion problem is ultimately the same for both classes of vessels.

The average operating engineer on a merchant vessel is a very busy and versatile person. Though ventilation is being given much more consideration, it is still a relatively minor portion of the operating engineer's responsibilities. For this reason simple, fool-proof and tamper-proof designs are essential. Adequate label plates and operating instruction are considered an integral part of the job.

Insulation has been and still is a problem on merchant vessels. Much has been and will be learned from the Navy's studies. Frequently, quarters ventilation systems have been considered inadequate when the insulation of the engine room boundaries and equipment was actually at fault. Poorly applied insulation, irrespective of its good basic qualities, is useless because it will not withstand vibration, shock, and general abuse of the service.

Heating arrangements on merchant vessels are in general the same as those used by the Navy. Hot blast heating is generally provided where mechanical supply is available and direct radiation is used in other spaces requiring heat. Unit heaters are used to a very limited extent. In merchant work volume dampers on supply outlets are used to a greater degree for regulating individual room temperatures. Today this is more common than usual because of the necessity for conserving critical materials and labor.

Natural ventilation is relied on more extensively on merchant vessels. The effectiveness of such ventilation, however, has been materially reduced in many cases by the addition of light-proof features. The skylights commonly employed for exhausting galleys and machinery spaces are notable examples. Mechanical exhaust is now specified for galley spaces, but because of the lack of generator capacity this is not practical for machinery spaces. Also, in many cases, the deck space previously available for natural ventilators, is now occupied by guns and mechanical ventilation must now be used if desirable conditions are to be maintained.

Under normal operation merchant vessels, particularly those engaged in passenger trade, can be designed to suit the weather conditions which they are most likely to encounter. However, at present designs must be universal, so that reasonable comfort may be achieved at all times.

Ventilation and heating must satisfy the human element both on naval and merchant vessels. Labor organizations have been interested in the improvement of conditions on merchant vessels. One instance is that bracket fans are required in all crew's spaces. Unfamiliarity with the limits of comfort which a ventilation system can achieve has frequently caused complaints and in many instances room temperatures lower than that outside are expected. A few of the early cargo vessels were provided with a water-spray humidifier for the heating system. We received a telegram from the crew of one of these vessels who were trying to determine whether the air conditioning system was out of order or whether the operators were too cheap to operate it during hot weather. Most of the unlicensed crew serve but a short time on each vessel. In shifting from one vessel to another they have the opportunity to judge for themselves which vessels are most comfortable. The Liberty Ships, which were provided with only natural ventilation, in order to simplify construction, have been criticized by many of the personnel serving on them, who are accustomed to sailing on ships fitted with mechanical ventilation.

The ventilation treatment provided on merchant ships is similar to that of the Navy, except for several items:

1. Living quarters usually consist of staterooms housing one to three persons. Group berthing is used only on transports.
2. Working and heat producing spaces are not so compact.
3. Stowage spaces, except cargo holds, are close to heat producing spaces. Also, all kinds of material are carried in cargo hold spaces. Some cargo hold spaces are adjacent to the engine and boiler room and are bounded by exposed uninsulated steel structures.

Living spaces on merchant vessels such as staterooms, offices, and messrooms are usually provided with filtered ventilation air, the volume being determined by three separate considerations: air change, temperature rise, number of occupants. It should be noted that where zone heating is used, the ventilation volume must be balanced against the heating requirements. These spaces are exhausted to passageways and adjacent toilet and shower spaces by means of louvered doors. Toilet and shower spaces are provided with mechanical exhaust, while passageways are provided with mechanical or natural exhaust depending on their location and relation to heat producing spaces. Registers are frequently used as well as diffusers because spaces are not so confined. Outlet velocities are somewhat higher than those used on land as fan motors are operated at reduced speed during the heating cycle.

The mechanical air supply for machinery spaces is distributed to the watch stations by means of directional terminals. Mechanical exhaust is used to a limited extent, while natural ventilation through skylights is most common. While machinery spaces are definitely a problem and considerably more research must be made to take full advantage of the ventilation provided, the situation is not so serious on merchant vessels as it is on Naval vessels. This is principally due to the fact that less compact arrangements exist and combustion air is taken directly from the machinery space, except on some Diesel vessels. Also, there are fewer watch stations on a merchant vessel and the steam pressures and superheat are less.

Many designers have labored under the misapprehension that machinery space ventilation is provided only for the benefit of the men stationed therein and the equipment housed therein. Also they have relied on the insulation, faulty in many cases, to prevent heat leakage through machinery space boundaries common to living quarters. For these reasons many have attempted to correct bad conditions by providing an abundance of ventilation to the living quarters surrounding these spaces. I believe the correct solution is to provide mechanical exhaust for the machinery spaces as described in the paper in order to eliminate leakage of hot air to living spaces and to prevent *hot pockets* within machinery space adjacent to living spaces.

On merchant vessels a much larger percentage of the ship's internal capacity is used for stowage purposes. The ship's stores occupy only a negligible portion of this volume. These are provided with mechanical supply in order to keep them at a reasonable temperature. Dry cargo spaces are treated in a similar manner for the same reason. This ventilation also serves to carry off moisture given up by the cargo and dunnage. Some vessels have been provided with dehumidifying systems which prevent the cargo from damage due to condensation, molding, tainting, etc.

Refrigerated spaces are divided into two classes on merchant vessels, ships refrigerated stores and refrigerated cargo spaces. The refrigerated stores are cooled by wall and ceiling coils, or cold diffusers depending on the temperature to be maintained.

Cargo spaces are cooled by the same methods, except that cold diffusers are not suitable because of the size of the space. Instead, a duct system is provided for creating air circulation and uniform temperatures. In some cases the recirculated air is cooled by blowing over pipe coils installed on the walls of the refrigerated space. In other cases a central cooling coil is provided for this purpose. The particular form of cooling system installed depends on the variation in types of cargoes handled, whether the cargo is received hot or precooled, and on other factors which normally enter into such designs ashore.

The authors cannot overemphasize the importance of balancing ventilation systems and the lack of consideration given this item in the field. The importance of considering the problem of balancing at the time the various systems are designed also

cannot be overemphasized because ideal conditions for balancing practically never exist on the job. The present rate of ship production as well as the great lack of competent personnel have undoubtedly aggravated this situation, but the basic reason is the general tendency for contractors to under-estimate the importance of ventilation as a whole. Misapplication of air measuring instruments, uncalibrated instruments, ignorance of recent research, and untrained personnel are among the chief direct offenders in this case.

At the present time, the Maritime Commission has not standardized on any ventilation equipment. Up to a short time ago all equipment and plans were submitted to a central office for approval. Recently, four regional offices were established in Philadelphia, Chicago, New Orleans and San Francisco and now handle most of this work. With the new set-up standardization may be desirable, but as yet no effort has been made in this direction.

Under the original long range program, the Commission did not purchase mechanical equipment. Several years ago an emergency program was initiated which operates more or less independently of the long range one. The program is devoted to the construction of a few types of vessels which are built in very large numbers, such as the Liberty Ships. Some of these types were originally designed by private shipyards and working plans are based on equipment produced by particular manufacturers. The Commission's practice is to let a contract for a number of duplicate vessels and furnish copies of the original working plans to the shipyard. Equipment, however, is generally purchased by the Commission through competitive bidding. I think that the advantages of standardizing ventilation and heating equipment are quite apparent with this procedure. Unless the original manufacturer also is awarded the bid, working plans will require modification to suit another manufacturer's equipment. Sometimes, the differences may not be detected until work has well advanced in the yard. In other cases major changes in arrangements may be necessary because of interferences and space limitations. Furthermore, many of the yards constructing these vessels do not have adequate personnel for modifying working plans. Standardization may also serve a useful purpose in our long range program, for with the present speed of production detail plans of equipment are often submitted after it (the equipment) has been shipped to the yard. The original purpose for which the plans are prepared is therefore voided. Standardization tends to eliminate the possibility of inferior, unsuitable equipment competing with superior specially designed equipment. We must acknowledge that the "know how" is worthy of compensation. However, if the standard is specific, the difference in spread between competitive bids will be more rational, thus indicating those which are too low to be within reason, as well as those which are abnormally high.

The ships of both the Navy and Maritime Commission have at least one requirement in common,—to efficiently carry materials on water. Every pound of unnecessary weight carried reduces the utility of the vessel by just that much. The Navy expresses this utility as fighting power, while the Merchant Marine expresses it as earning power.

Today there is no competition in the shipping industry. After the present conflict, our merchant marine will have to compete with both domestic and foreign air transportation, as well as the ships of other nations. We know that efficient design and operation is the key to success in any business. Also, higher costs of production and standards of living in this country must be overcome by producing more efficient vessels than our competitors. Our records show that the faster long range program vessels can compete with the relatively slow cargo vessels such as our Liberty Ships. Here again more efficient design helps overcome the enormous penalties of higher speeds.

While at present, air conditioning is virtually non-existent on merchant vessels, ships of the future will rely considerably on this method of comfort to attract the passenger trade. Comfort cooling both on ships and ashore must satisfy the human

element. However, psychrometric conditions, operational features and other factors peculiar to ships must be thoroughly considered if a practical solution is to be obtained which produces something more than advertising value. Efficient, simple and reliable equipment, designed particularly for the service, will play a large part in the general acceptance and success of comfort cooling in the future.

It would not be fair to close such a discussion as this without noting a few of the meritorious advancements to our particular fields of interests which the Navy Department has fostered.

The Navy's early interest in air conditioning is illustrated by the following quotation from *Fifty Years of Naval Engineering in Retrospect,* by Herbert M. Newhaus, which appeared in the February, 1938 issue of the *Journal of the American Society of Naval Engineers:*

> "That these Naval Officers (Bureau of Steam Engineering) participated in other than purely Naval Engineering is illustrated by the installation of an air conditioning system in the White House during the illness of President Garfield (1881). Here filtered air from blowers was passed over ice into the sick room much as in some present day installations. Humidity was even then recognized as the principal and most disturbing factor in air conditioning."

In 1890 the Navy found that few reliable data were available pertaining to the performance of fans and the laws governing air flow in ducts. Therefore according to form, the Navy started to find out for itself, the first results being presented before the *Society of Naval Architects and Marine Engineers* in 1905, by Admiral D. W. Taylor, U.S.N. This report included a complete study of the friction and static pressure losses in pipes, elbows, transitions, etc. It also introduced a new method of testing fans which is very similar to that in use today. In addition, this report introduced backward curved wheels for centrifugal fans in an effort to prove that the commercial fans then available were not suitable for Navy use. Needless to say Admiral Taylor's report met with considerable opposition. One example worth while quoting reads as follows:

> "I have estimated that the entire amount of fans required for Marine Service in the course of a year would only constitute about 3 per cent of the business of one of our large fan manufacturers, and the request of the Navy Department seems to be about the only *urgent* one for a fan of greater efficiency. Furthermore, the demands of the Navy Department for an absolutely convertible fan is of great hindrance to the production of a fan of increased efficiency."

Time alone has proven the value of Admiral Taylor's report to the industry, as well as the Navy. Many years have passed since then and many other contributions have been made. One of the most recent developments is the axial flow fans which this paper has modestly referred to.

The Navy has learned that every piece of equipment must pay its own way. An expression of Admiral G. W. Melville, appropriately illustrates this point:

> "That the location of forced draft blowers is a matter of serious importance. In some of our ships, due to the demand for all other space for other purposes, the blowers had to be located in corners or pockets in the fire room, where it was impossible for human beings to give proper attention, owing to the intense heat due to lack of ventilation. In the "Cincinnati" temperatures as high as 205 degrees F. were noted and the Commanding Officer when investigating the case, personally had his face scorched.
>
> "The blowers must be placed where they can be properly cared for or else they are useless and might as well be left ashore."

The last sentence can be considered a general law in both naval and merchant practice.

The following statement may not be appropriate, but it tends to prove that there is very little new under the sun.

> "That no oil fuel installation should be permitted for marine purposes which would not permit renewal within twenty-four hours of all grates and bearing bars, so that a return to coal could be accomplished within a reasonable time in case of failure of oil supply."

The foregoing quotation was taken from a report published by the Bureau of Steam Engineering in 1902.

1225

No. 1225

SOME ENGINEERING PROBLEMS OF THE NEW VEGETABLE DEHYDRATION INDUSTRY

By W. B. Van Arsdel,* Albany, Calif.

DURING the past year a renaissance of the vegetable dehydration industry has brought into being scores of new dehydration systems and a large number of more or less radical modifications of conventional types of dehydraters. A good many processes conceived during the great dehydration boom of 25 years ago are now being resurrected; once again there is a market for novel ideas. There are many concerns that now wish to install dehydraters, but have only vague notions about what a good dehydrater should be. Almost any system, no matter how fantastic, may be reasonably sure of a respectful hearing.

Heating and ventilating engineers know that there are exact physical laws which describe accurately the operation of every dehydrater. These laws may be complex, but there is no mystery about the conservation of mass and of energy and the second law of thermodynamics, which rule in this sphere as they do in all physical processes. Any competent engineer can, by the use of these laws and by application of his knowledge of the properties of water and of air, calculate what the top performance of any particular dehydrater can possibly be. No more than that is needed to deflate the pretensions of some of the more inept of the current dehydrater designs.

Much more is needed at the present juncture, however, than the mere weeding out of mistaken or extravagant systems. The whole matter of dehydrater design and construction is in a state of flux and rapid development, and there is great need for the immediate application of the highest type of engineering talent to this problem.

Complete Plant Design Needed

In the first place, much more attention needs to be paid to well-rounded and complete plant design. It should not be forgotten that the dehydrater is only one piece of equipment in a production line which must work as an integrated whole. In some important respects it would be preferable to tailor the dehydrater to fit the requirements of the rest of the operation, rather than to build the whole plant around the dehydrater. Today there are very few engineering firms, consultants, or equipment concerns that confess to broad enough competence to undertake this integrated plant design. And yet it is certain that within the coming year or two there will be hundreds of plants in the United States which will need that kind of service. Without it there will surely be numerous cases of serious unbalance, waste of equipment and materials, and failure to reach production goals.

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Within the more narrow field of the design of the dehydrater itself a great deal needs to be done, and time presses inexorably. In spite of the great amount of high grade work done in the past, as evidenced by hundreds of publications and patents, two circumstances have reopened the whole subject. One is the impact of a wartime scarcity of materials and equipment, the other is the appearance of much new knowledge about the manner in which various vegetables dry and the effect of processing methods on the quality of the product.

WARTIME ADAPTATIONS

The economic consideration is substantially different from what it would be in time of peace. Equipment cost must now be weighed with less regard to dollars and more regard to pounds of steel and man-hours of skilled fabricating labor. One consequence is an intensive drive to adapt existing dehydraters of many types to this new job. Some converted fruit dehydraters are performing very well. Some others are relatively ineffective. There is room for high class engineering in the modification of such equipment, and in the adaptation of numerous driers in other industries whose operations have been curtailed because of the war. Another consequence is that great emphasis must be placed on such rational design of dehydraters as will assure the absolute maximum of output from each unit consistent with high quality of product. This demand has led to the rapid development of multi-stage dehydration processes in which the equipment in each stage is matched with the drying characteristics of the material. This development will be discussed at greater length later in this paper.

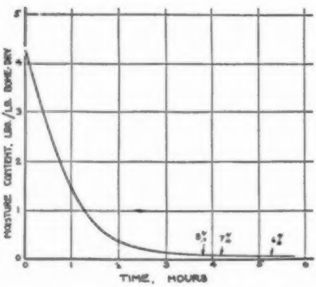

FIG. 1. TYPICAL VEGETABLE DRYING CURVE

Russet potato, blanched 5/32-in. strips, wood-slat tray, load 1.5 lb per square foot, cross-circulation, air velocity 500 fpm, air temperature 150 F, wet-bulb temperature 90 F.

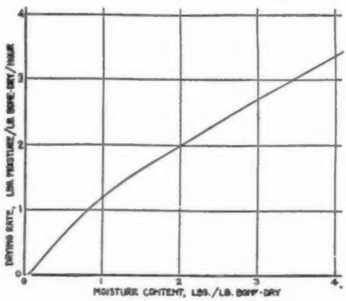

FIG. 2. DRYING RATE CURVE

Russet potato, blanched 5/32-in. strips, wood-slat tray, load 1.5 lb per square foot, cross-circulation, air velocity 500 fpm, air temperature 150 F, wet-bulb temperature 90 F.

Some of the other traditional economic yardsticks suffer a change of scale because of the war. Local circumstances of fuel shortage or scarcity of labor may have a powerful effect on design. If fuel supply is not a serious bottleneck, the pressure for production is likely to make fuel economy unimportant. The same thing is true of power usage.

NEW DATA ON VEGETABLE DEHYDRATION

During the past two years a great body of specific knowledge about vegetable dehydration has been built up. This knowledge may be divided into two distinct, but interrelated, fields—information about drying rates and information about the effects of processing methods on product quality.

Past work on the design of vegetable dehydraters has had to be conducted without benefit of exact data on the drying rates of vegetables. This information is rapidly becoming available. As anyone who has ever tried to design a dehydrater knows, the general physical laws of thermodynamics and of air circulation are only a part of the information needed—a necessary part, but not sufficient. In addition the designer needs to know how fast the moisture will evaporate from the product, and what the relation of that rate

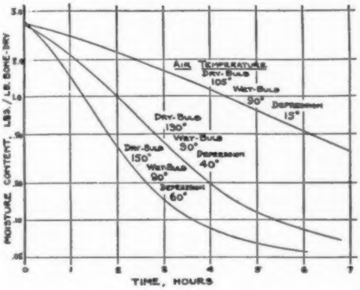

FIG. 3. DRYING CURVES AT DIFFERENT WET-BULB DEPRESSIONS

Russet potato, blanched 5/32-in. strips, wood-slat tray, load 1.5 lb per square foot, cross-circulation, air velocity 500 fpm.

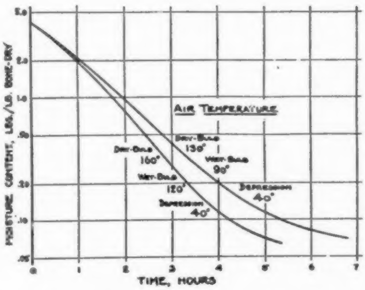

FIG. 4. DRYING CURVES AT DIFFERENT TEMPERATURE LEVELS

Russet potato, blanched 5/32-in. strips, wood-slat tray, load 1.5 lb per square foot, cross-circulation, air velocity 500 fpm.

is to all of the pertinent variables. It is only as he applies such knowledge that he is able to make really effective use of the materials and equipment at his disposal.

The evaporation of water from a mass of cut pieces of vegetable involves a complex train of phenomena. Within the pieces moisture must diffuse through cellular tissues which change in properties as the drying progresses. Sugars and other soluble substances become more and more concentrated, the vapor pressure of the solution falls, and its viscosity rises. Shrinkage in dimensions is high and non-uniform. The situation is so complicated that theoretical analysis has not been very helpful and it has been necessary to depend almost entirely on experimental methods which simulate commercial conditions as closely as possible.

DRYING CHARACTERISTICS OF VEGETABLES

If drying conditions are maintained constant, the loss of weight of a trayful of cut vegetable pieces follows some such course as the curve shown in Fig. 1. This curve is typical in two respects, namely, the very rapid initial rate of

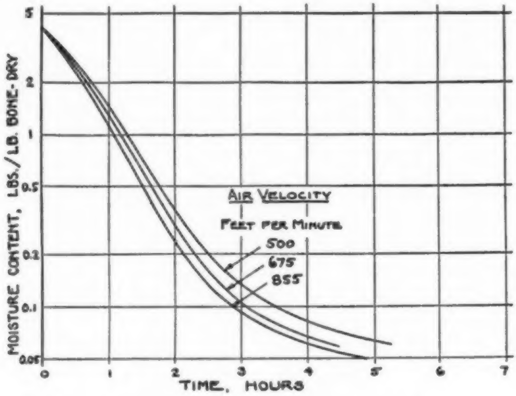

FIG. 5. DRYING CURVES AT DIFFERENT AIR VELOCITIES
Russet potato, blanched 5/32-in. strips, wood-slat tray, cross-circulation, air velocity 500 fpm, air temperature 150 F, wet-bulb temperature 90 F.

loss of moisture, and the very slow final approach toward an equilibrium moisture content. All of the experimental results obtained with the common vegetables have exhibited these same characteristics. Any dehydration system which fails to take these two characteristics into account fails, except accidentally, to attain real effectiveness.

Data on drying rates have commonly been analyzed by plotting the instantaneous rate of drying against the moisture content. Fig. 2 shows the curve

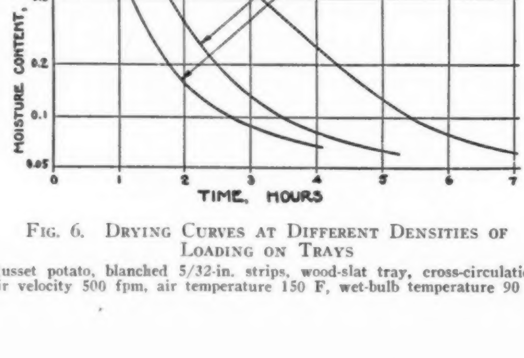

FIG. 6. DRYING CURVES AT DIFFERENT DENSITIES OF LOADING ON TRAYS
Russet potato, blanched 5/32-in. strips, wood-slat tray, cross-circulation, air velocity 500 fpm, air temperature 150 F, wet-bulb temperature 90 F.

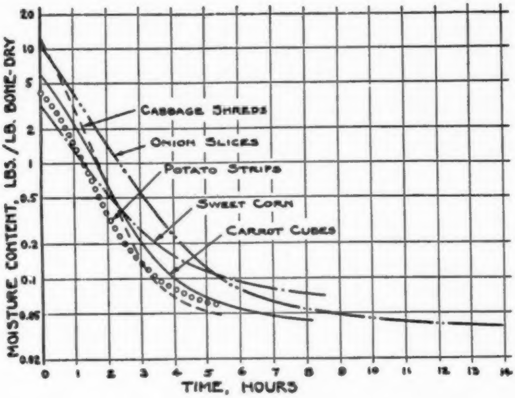

FIG. 7. DRYING CURVES FOR SEVERAL COMMON VEGETABLES

Russet potato strips, Chantenay carrot cubes, Cannonball cabbage shreds, sweet corn, Yellow Globe onions. Varying loads per square foot of tray. Air velocity 500 fpm, air temperature 150 F (onions 140 F), wet-bulb temperature 90 F.

which results from that method of handling the same date given in Fig. 1. The rate of drying falls continuously as moisture content falls. Contrary to experience in the drying of some other wet materials, there is no phase during which the drying rate remains constant, and the curve cannot be expressed at all accurately by one or two straight lines. Intensive study of a large number of drying curves of this kind has led us to the belief that they are not well adapted to the formulation of such general expressions for dry-

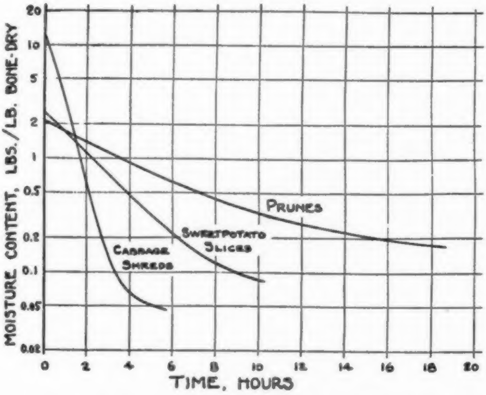

FIG. 8. DRYING CURVES FOR A STONE FRUIT AND TWO VEGETABLES

ing rate as the designer really needs. The main difficulty is that the precision of estimate of drying time from such a curve falls rapidly in the lower part of the curve. Since most of the time required for drying will be consumed at these low drying rates, the lack of precision in the estimate may be serious.

It has been found more profitable to correlate such data as those of Fig. 1 directly, instead of using the instantaneous rates. Even at best, this correlation is a complex procedure, involving the separation of at least nine independent variables.[1] Only a progress report on the work can be made at this time. As promptly as possible, completed segments of the investigation will be published.

A comparison of a group of drying curves in which the independent parameter is wet-bulb temperature depression is shown in Fig. 3. The curves are straightened out materially by plotting on semi-logarithmic paper. As might be expected, more rapid drying is associated with higher wet-bulb depression.

In Fig. 4 a comparison is given between two drying curves in which wet-bulb depressions are equal but temperature levels are different. Drying is more rapid at the higher temperature level.

A group of curves in which the only difference is air velocity across the tray on which the material was spread is compared in Fig. 5. Drying is more rapid at the high air velocity, but the effect is confined almost entirely to the upper portions of the curves, that is, the high-moisture portion of the run.

Fig. 6 compares curves from three runs in which the weight of loading of the moist material on a unit area of the tray was varied. The retarding effect of heavy loading is apparent.

A comparison is shown in Fig. 7 of drying curves on several different vegetables, all under approximately the same drying conditions.

The degree of difference in drying rate between a typical fast-drying vegetable (cabbage), a slow-drying one (sweet-potatoes), and a stone fruit (prunes)[2] is indicated in Fig. 8.

We have been expressing our correlations of the effects of different variables both through empirical equations and through nomographs. A nomograph has advantages both in the ease of its use and in the fact that its scales may be drawn only as far as the experimental data themselves go, so that unwarranted extrapolation is discouraged. Fig. 9 is one such nomograph, still incomplete in the important region of low moisture content.

EFFECTS OF DRYING CONDITIONS ON QUALITY

Our knowledge of the effect of drying conditions upon the quality of the dry product is also much more comprehensive than it was two years ago.[3] While investigations are still far from complete, for these relations are much more complex than those involved simply in the rate of drying, certain generalizations may be made with confidence:

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[1] Variety of vegetable, method of preparation, shape and size of pieces, thickness of layer, type of tray or other support, mode of exposure to the air stream, temperature of air, humidity of air, velocity of air, and others of less importance.

[2] Taken from Some Developments in Fruit Dehydration, by Rene Guillou; Agricultural Engineering, November, 1942.

[3] *Quality* comprises such factors as palatability, appearance, nutritive value, and stability during storage.

1. Rapid dehydration favors high retention of product quality.

2. Damage to quality is a composite effect of time and temperature; the coefficient of the temperature effect appears to be considerably higher than the doubling in each 10 C which characterizes many chemical reactions, so that there is a decided appearance of a limiting, or *critical,* temperature for safe operation. The important temperature is, of course, that of the material itself, not that of the air. There is some evidence that temperature damage may occur during any stage of the drying. The *critical* temperature of damage varies with the variety, maturity, length of storage, method of preparation, and possibly other factors.

3. Loss of quality during storage after dehydration is highly dependent on moisture content. The quality of a vegetable stored at 2 per cent moisture for 6 months may

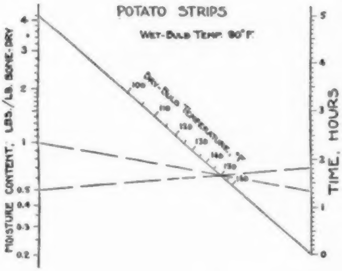

FIG. 9. PARTIAL NOMOGRAPH FOR DRYING TIME

Russet potato strips, wood-slat tray, cross-circulation, air velocity 500 fpm.

be substantially higher than that of one which is dehydrated only to 5 per cent moisture and stored for the same length of time, although the latter may be better when freshly dehydrated.

SPECIFICATIONS FOR WARTIME DEHYDRATION SYSTEMS

Summarizing what we now know about vegetable dehydration, it appears that the following characteristics should be incorporated in a dehydration system which is to fit the realities and the urgencies of our wartime needs:

1. *It must produce a product of the highest quality consistent with large-scale practicability:* One consequence of this requirement is that the drying potential must be maintained as high as is safe and practicable throughout the process. High air temperature is desirable so long as the product maintains itself at a much lower temperature through evaporative cooling. A careful balance must be struck between damage caused by high product temperature and damage caused by slow drying.

2. *It must permit a low final moisture content to be attained:* In order to meet this requirement the product must be finished under conditions of high drying potential.

3. *It must give substantially uniform treatment to all pieces of the product:* High quality cannot be expected if the product consists of a heterogeneous mixture of scorched and underdried pieces.

4. *It must turn out maximum output per unit of scarce metals used in construction:* While there is no precise way to equate different scarce materials and equipment, the principle must be followed faithfully. Too often it comes into collision with an engineer's natural desire to build a substantial permanent unit which will be a credit to him in peacetime. The principle should apply to scarce skilled labor as well as to materials. Common sense suggests, however, that shoddy construction is no more justified in wartime than at any other time. Freedom from breakdowns and from expensive repairs and replacements may, indeed, be more important now than normally.

The pressure for high output per unit of equipment is a force which favors multi-stage dehydration. A comparatively small predrier can evaporate three-quarters or more of the total moisture very rapidly. The last few per cent of moisture may be evaporated in a bin type of finishing drier. The main drier can thus be liberated to do only the part of the job it can handle most effectively. The really good dehydration system will use a type of equipment at each stage which corresponds to the properties of the material at that stage.

5. *It must not require excessive operating labor:* Wartime manpower shortages will become increasingly serious. The proper balance between mechanization, using more metals and machine work, and pure hand operation, using more labor, will be difficult to strike justly; it may vary from one plant to the next. In some types of dehydraters no amount of labor can obviate the need for automatic control instruments. The shortage of such instruments is so acute, however, that the design should, if possible from other standpoints, be based on a type which requires only a minimum of automatic control. The designer must bear in mind that simplicity of control is going to be essential during the next few years, because dehydrater operators will often be only semi-skilled men or women.

6. *Control devices must be flexible and precise:* Vegetable dehydration by present standards demands accurate control of drying conditions. In the case of some vegetables the margin between desirable rapid drying and scorching is very narrow. Flexibility is essential not only so that the necessary compensations can be made for changing weather conditions, but also because most dehydraters must handle any one of several vegetables satisfactorily.

SUGGESTIONS TO DEHYDRATER DESIGNERS

These wartime requirements open a wide field to the designer of dehydraters. In particular, the multi-stage process offers almost limitless possibilities. Methods that would be open to serious doubt as to utility or economy if used for the entire drying process may fit naturally into one stage of a composite operation.

In the past there has been a strong tendency for designers to work out a single design, possibly to patent some feature of it, and then concentrate on that one design, and that one only, for every job. Equipment concerns have come to be known for their own characteristic types of driers. There are now scores of these competing *systems*. Most of them are advertised in terms which suggest that they are panaceas. We think the fact is that most plants need a tailor-made job, not a ready-made off the rack. We should like to see the equipment concerns undertake to design and build for a plant whatever type of dehydrater best fits its actual needs. If the proper equipment for a certain plant is a continuous-belt predrier, then a tunnel, and finally a vacuum finisher, then there should be engineering firms with enough catholicity of outlook, enough breadth and tolerance, to prescribe just that and not to insist that it shall be called somebody's *system*.

Finally, we must warn about the uselessness of designing dehydraters without a real knowledge of the way a modern dehydration plant operates. No matter how right the design is from a theoretical standpoint, it is wrong if it

does not fit smoothly into the necessary framework of operations in a plant. There is no substitute for actual knowledge of these operations. Firms and consultants in this field are, however, urged to familiarize themselves also with the technological work in dehydration which is being carried on in U. S. Department of Agriculture laboratories. Since a large part of this work is still unpublished, personal visits will be found to be much more satisfactory than correspondence.

DISCUSSION

H. E. LEWIS, Toledo, Ohio: I would like to ask if there has been any effort made to record the different drying times and pickup (or elimination) of moisture on the basis of vapor pressure differentials, rather than by the selection of wet- and dry-bulb temperatures?

MR. VAN ARSDEL: We have made numerous attempts to correlate the figures between different drying runs on the basis of two or three possible ways of expressing vapor pressure deficit. We have finally come down to correlation on the basis of wet- and dry-bulb temperatures as being not only the most direct but as giving just about as good correlation as anything else we have found. The conditions under which vegetables are dried commercially are so complex that complete theoretical analysis is impractical. Use of a physical concept like vapor pressure deficit therefore offers no particular advantage.

CYRIL TASKER, Toronto, Ont., Canada: I would like to ask whether there has been any use yet of chemical dehumidification to produce very dry air with which to finish off the drying process.

MR. VAN ARSDEL: Yes, sir; there is growing use of air desiccants in the finishing operation. Silica gel has been used; activated alumina and lithium chloride also might be considered. That is only necessary in the finishing operation itself to remove the last few per cent of moisture.

L. T. AVERY, Cleveland, Ohio: Following that point, what is the probable moisture regain after you have dried it?

MR. VAN ARSDEL: The material is packed in hermetically sealed cans, just as fast as possible after it is finished, and then, of course, there is no possibility of regain until it is opened up ready for use in the Army kitchen. Regain is a problem only if the product is packed in a container which will transmit water vapor.

J. D. SLEMMONS, Columbus, Ohio: I would like to ask the author if there is a possibility in the future of application of drying to farm products by the farm wife, something which has been done and which did not necessarily give the best quality.

MR. VAN ARSDEL: The question concerns the farm drying or home drying of products on a relatively small scale. Considerable work has been done along that line, and it is possible to get a reasonably good product that way. I am afraid that such products would be uneven in quality. Many of them would not pass present government purchase specifications, particularly as to their storage possibilities; but it would make a food product of some sort from vegetables which might be lost otherwise. Several of the State University extension services have information on this subject which they will send to inquirers.

No. 1226

THE PERFORMANCE OF SIDE OUTLETS ON HORIZONTAL DUCTS

By D. W. Nelson * and G. E. Smedberg,** Madison, Wis.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of Wisconsin.

Introduction

HORIZONTAL ducts with side outlets as used in heating and ventilating installations must have the flow of air controlled to give the desired effect. The performance of three sizes of side outlets on a horizontal duct equipped with three types of corrective devices and various extension lengths has been investigated. The results obtained should be of assistance in designing effective outlets to the end that distribution in rooms may be satisfactory.

Description of Apparatus

The test set-up used is shown in Figs. 1 and 2. The radial fan was connected by means of a suitable reducing section to an 18 in. x 18 in. duct 12 ft long. At the end of this 12-ft section a 7.55 in. nozzle was inserted. Downstream of the nozzle was a 7 ft length of 18 in. x 18 in. duct, in which were located adjustable and stationary static and total pressure tubes. The adjustable tubes were used in the calibration of the nozzle and the stationary tubes for setting the velocity during the various tests. This 7 ft length of 18 in. x 18 in. duct was followed by a 6 in. x 20 in. section of duct 15 ft long. Two openings were located in the side of this 6 in. x 20 in., and a third opening was located at the end of the duct. These openings were designed to permit the insertion of plates with the various sizes and types of outlets. The plates with the outlets had 1 in. extensions and were designed to permit the attachment of 2 in., 4 in., 6 in., and 12 in. long extension pieces resulting in total extension lengths of 1 in., 3 in., 5 in., 7 in., and 13 in. Three sizes of outlets were studied; namely, 3 in. x 10 in., 4 in. x 9 in., and 6 in. x 6 in. The static pressures in the system were measured by means of piezo meter rings and static pressure tubes at the points numbered 1 to 12 in Fig. 2. The dry- and wet-bulb temperatures of the air flowing in the duct were measured by means of suitable thermometers placed at the entrance to the 6 in. x 20 in. duct.

The volume of air flowing in the system was controlled by a damper and atmospheric relief door in the duct adjacent to the fan outlet and an adjustable cone on the inlet of the fan.

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** Research Fellow, University of Wisconsin, 1940-1942.

Presented at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1943.

Fig. 1. Photograph of Apparatus

Readings were taken using a deflecting vane anemometer with a duct jet at three stations in addition to flow measurements at the nozzle station. These stations were located before outlet No. 1, between outlet Nos. 1 and 2 and between outlet Nos. 2 and 3.

The three outlets were divided into 24 equal areas and the velocity at the center of each of these areas was measured. Angles of flow were measured at the center of each area for the middle row on each outlet.

Directive devices as shown in the lower right hand corner of Fig. 2 were designed to permit their convenient insertion at each of the side outlets. These

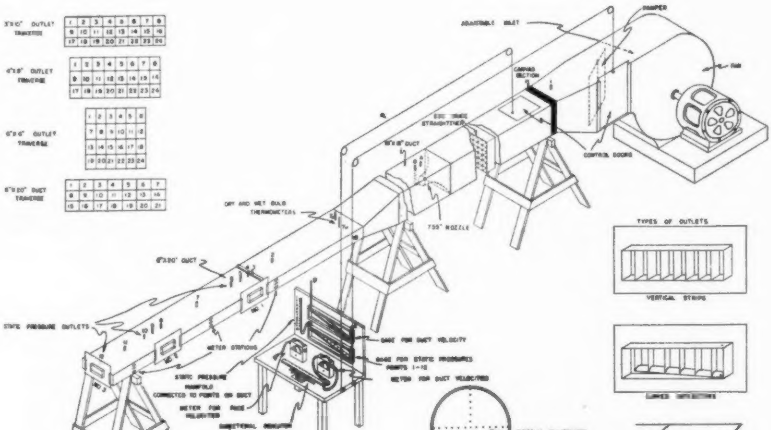


Fig. 2. Schematic Diagram of Apparatus

devices were not used at outlet No. 3 during the various tests, but the same length of extension was used on this outlet as on Nos. 1 and 2 in each test.

The test set-up was arranged so that all the static pressures could be read from a single point, and cables were arranged permitting the operation of the control dampers from the same location. Pressures were measured by differential or inclined manometers, depending on the range of the static pressures to be measured.

An air flow analyzer shown in Fig. 3 was used to study the flow pattern from the outlets. It has two glass plates spaced ¼ in. apart between which the model is placed, and between which the air flows, hence, it is limited to

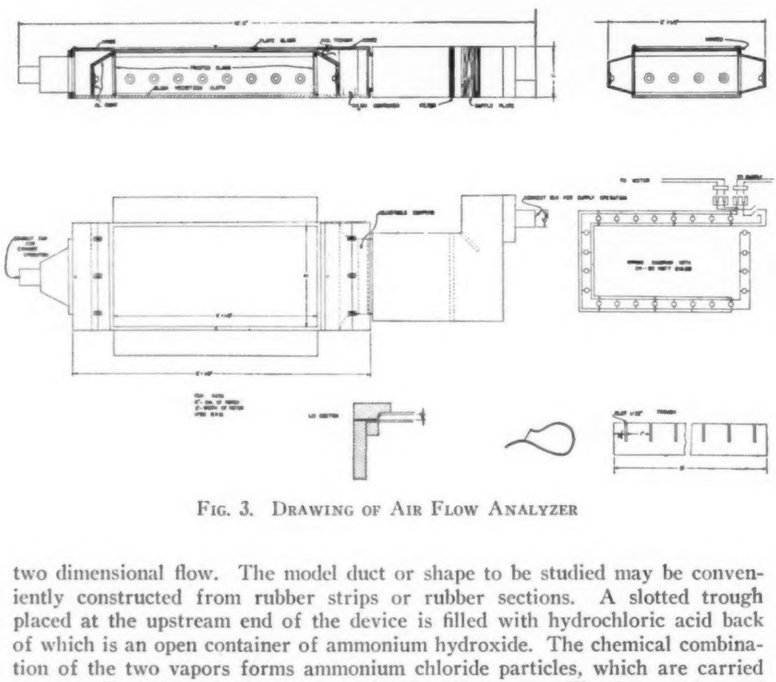


FIG. 3. DRAWING OF AIR FLOW ANALYZER

two dimensional flow. The model duct or shape to be studied may be conveniently constructed from rubber strips or rubber sections. A slotted trough placed at the upstream end of the device is filled with hydrochloric acid back of which is an open container of ammonium hydroxide. The chemical combination of the two vapors forms ammonium chloride particles, which are carried in streamlines through the apparatus by the air stream. The fan can be used at either end and when on the upstream side it acts as a supply fan, blowing the streamlines over the model. When used in this manner the plenum chamber on the upstream side is used. This chamber consists of a baffle arrangement which results in an even flow across the entire width of the apparatus. When used in this manner a vent strip is removed from the side of the apparatus allowing the streamlines to be projected outward from the opening in the duct work into the atmosphere. This method was used for the study of air flow in ducts since it was possible to discharge the smoke streamlines directly into the atmosphere which simulates actual duct operation. If the fan

is placed on the downstream side of the apparatus, it acts as an exhaust fan drawing the streamlines over the model. Operating in this manner the vent strip on the side is closed and the streamlines of smoke are drawn straight through the apparatus and discharged to the outside by suitable piping. This method of operation is used when studying the flow over various shaped objects such as wing sections and cross sections of condenser or boiler tubes, and reference to the literature shows that it has been used considerably in aeronautic studies.

The flow of air over various objects can be visually studied by means of this device, and a great deal of valuable knowledge thereby obtained. However, by recording the flow photographically the results can be studied at leisure and are of value for illustrating various points of theory. When the photographs for this paper were taken, the camera was mounted directly over the glass plates on a rigid support. The built-in lighting arrangement consisting of twenty-four 60-watt Mazda bulbs furnished sufficient illumination to allow a shutter speed of one-fiftieth to one-hundredth of a second with an aperature of *f*2.8. These speeds are based on the use of a film with a Weston tungsten rating of 64.

PROCEDURE

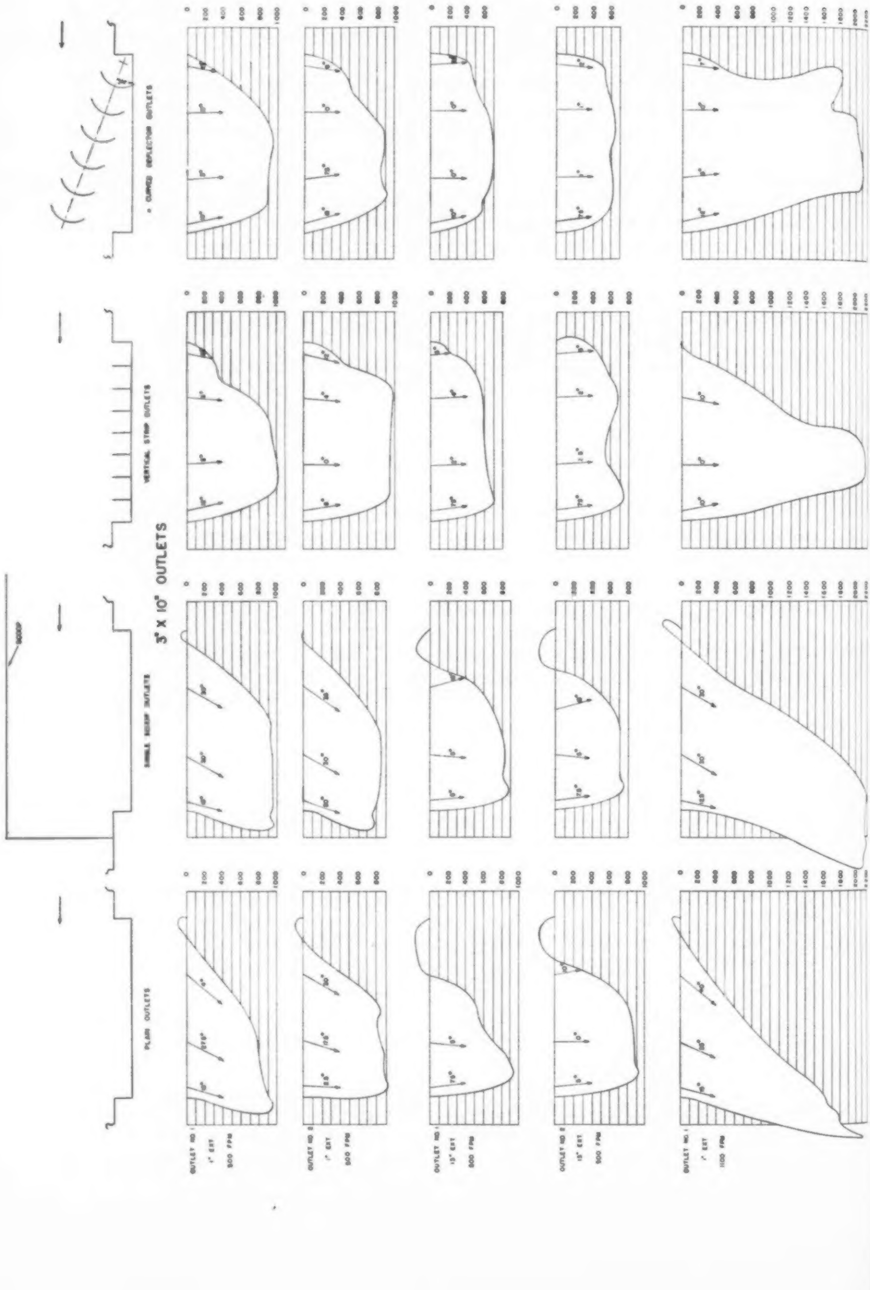
Duct velocities of 200, 500, 800, 1100, and 1400 fpm in the 6 in. x 20 in. section were maintained in the series of tests on various sizes and types of outlets and with various extension lengths. Tests were run at each of these velocities for the three sizes of outlets with 1 in. and 13 in. extensions. Only 1400 fpm was used for the intermediate lengths since preliminary tests indicated similar flow patterns were obtained as at lower velocities. These series of tests were carried out on the plain outlets, single scoop, vertical strip and curved deflectors in position in the outlets.

Calculations were made, previous to each test, of the correct gage setting at the nozzle corresponding to the velocity desired in the duct before outlet No. 1. In a test of each outlet combination, the direction and velocity at the center of each of twenty-four equal areas on each of the three outlet faces were determined by means of a deflecting vane anemometer used with a spot reading tip and a vane type angle measuring device termed a directional indicator.[1] In addition, the static pressures were read at the 12 points of the system shown in Fig. 2. The center velocity at the three measuring stations, where the deflecting vane anemometer was used with a duct jet, was also recorded during east test.

SUMMARY OF RESULTS

Figs. 4, 5, and 6 show the flow patterns at 500 and 1100 fpm for the three sizes of outlets with the various directive devices, and with 1 in. and 13 in. extension lengths. With the plain outlet and 1 in. extension, a very large angle of discharge was obtained. The addition of extensions reduced this angle. The same general flow patterns prevail when single scoops were used

[1] The Performance of Stack Heads Equipped with Grilles, by D. W. Nelson, D. H. Lamb, and G. E. Smedberg. See Fig. 2. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 279.)


3" X 10" OUTLETS
PLAIN OUTLETS
SINGLE SCOOP OUTLETS
VERTICAL STRIP OUTLETS
CURVED DEFLECTOR OUTLETS
SCOOP
OUTLET NO 1
1" EXT
500 FPM
OUTLET NO 2
1" EXT
500 FPM
OUTLET NO 1
15" EXT
500 FPM
OUTLET NO 2
15" EXT
500 FPM
OUTLET NO 1
1" EXT
1100 FPM

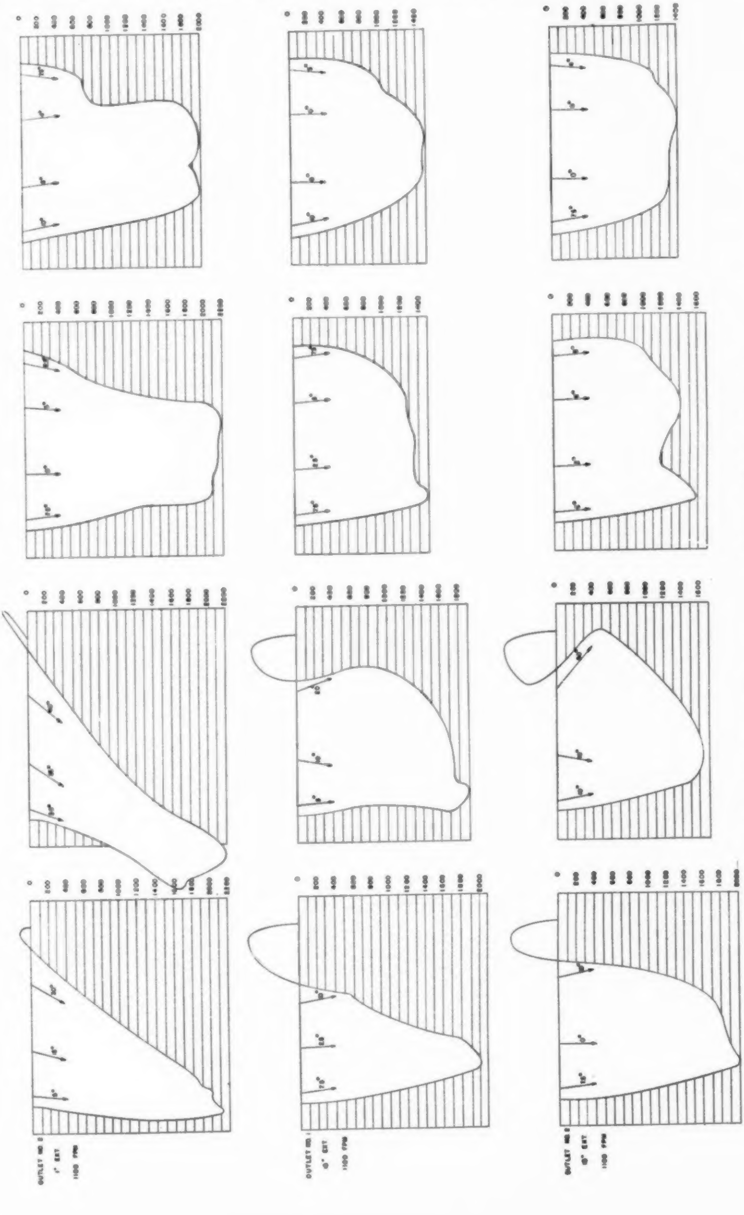
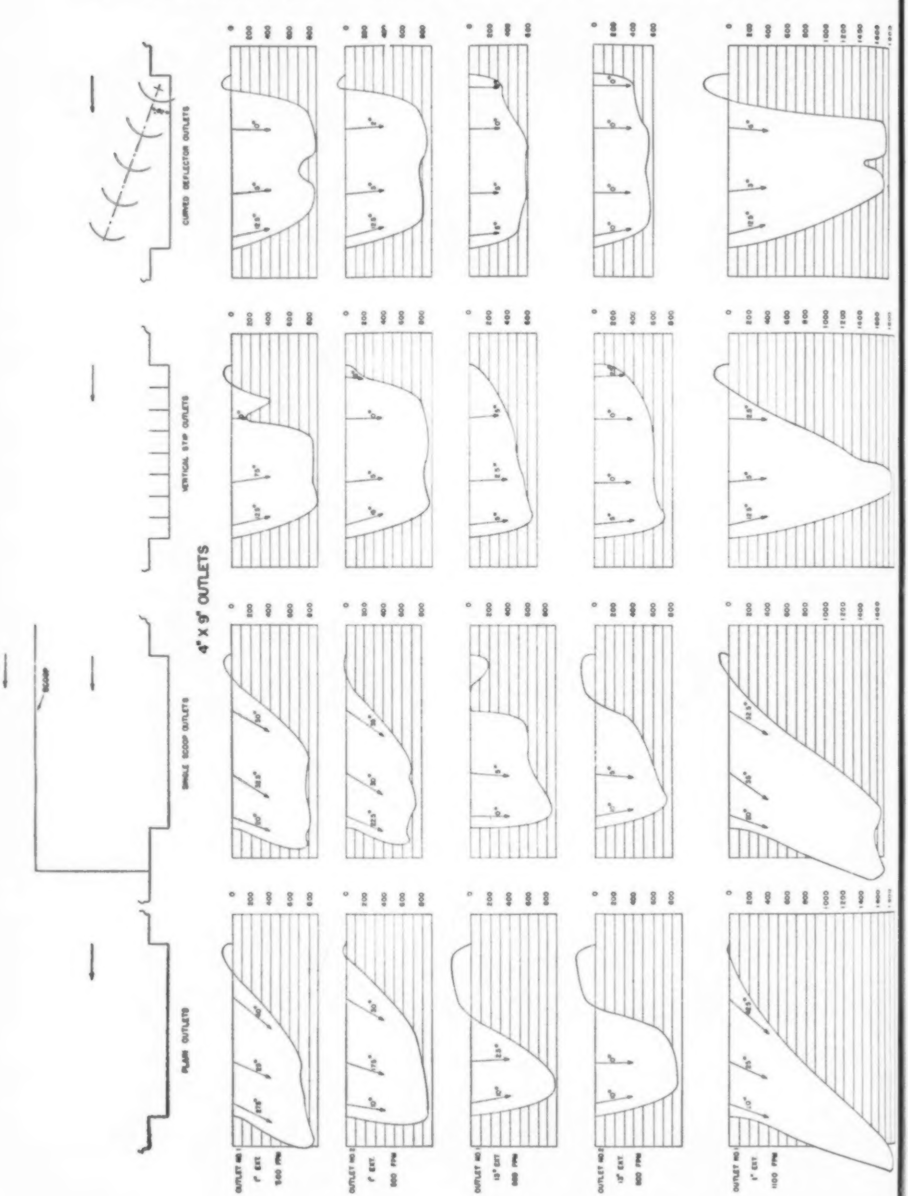


Fig. 4. Flow Patterns for 3-In. x 10-In. Outlet


4" X 9" OUTLETS
SCOOP

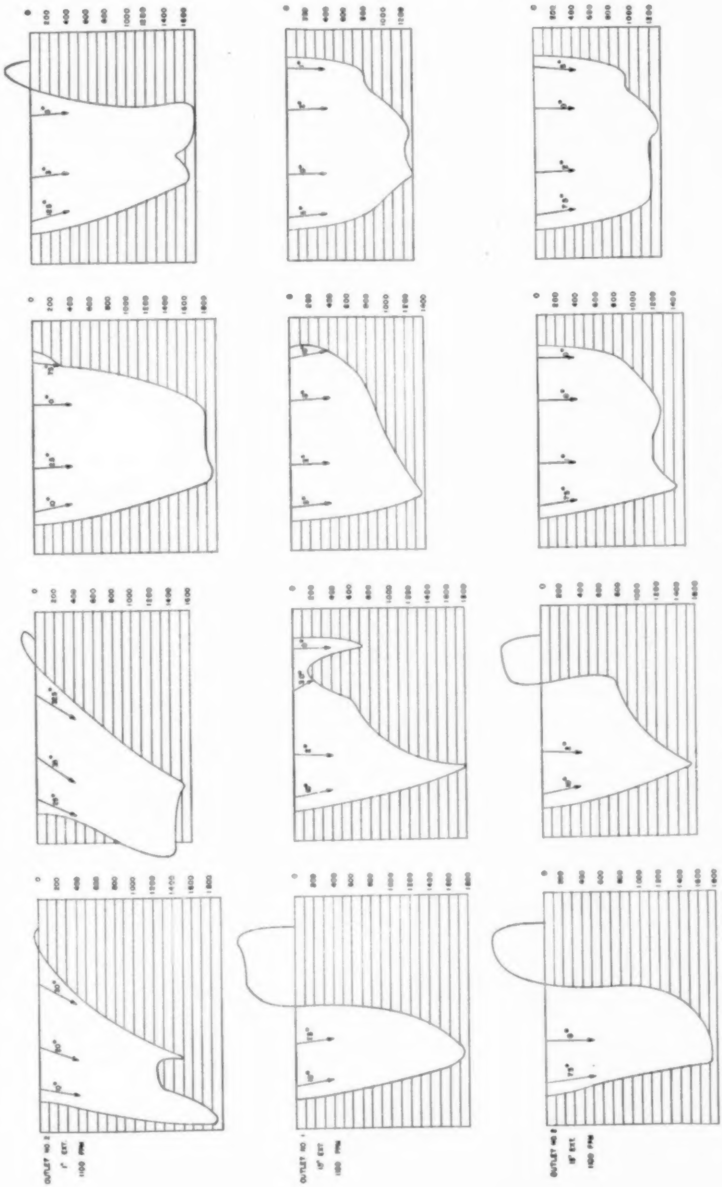
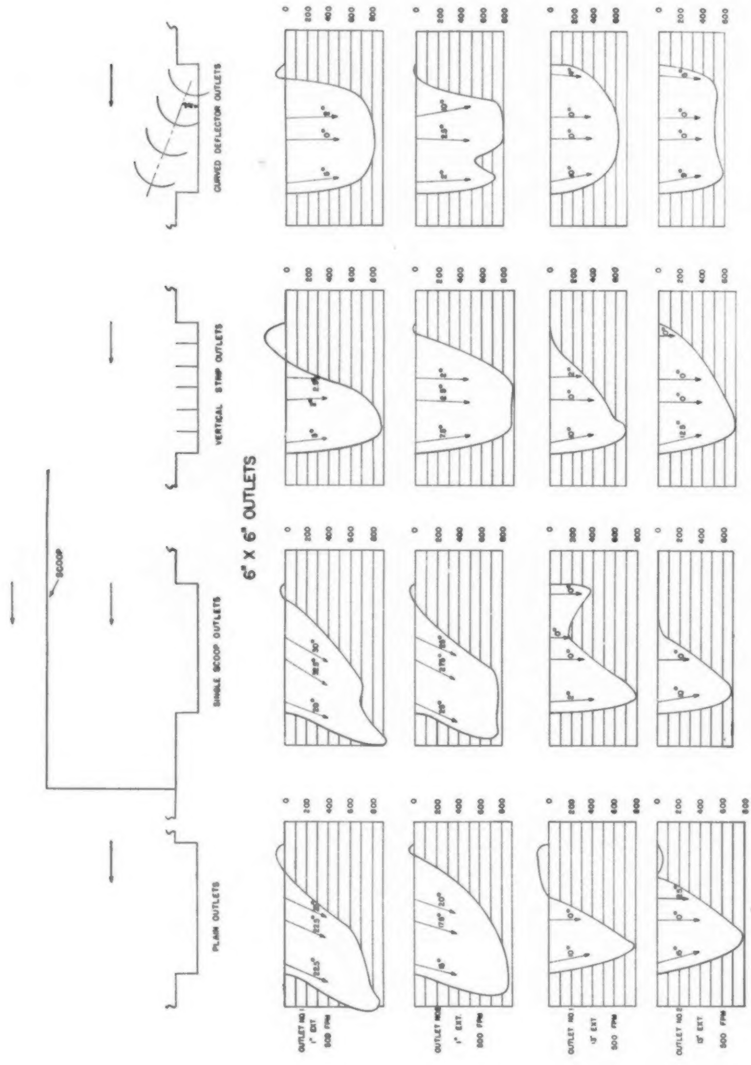
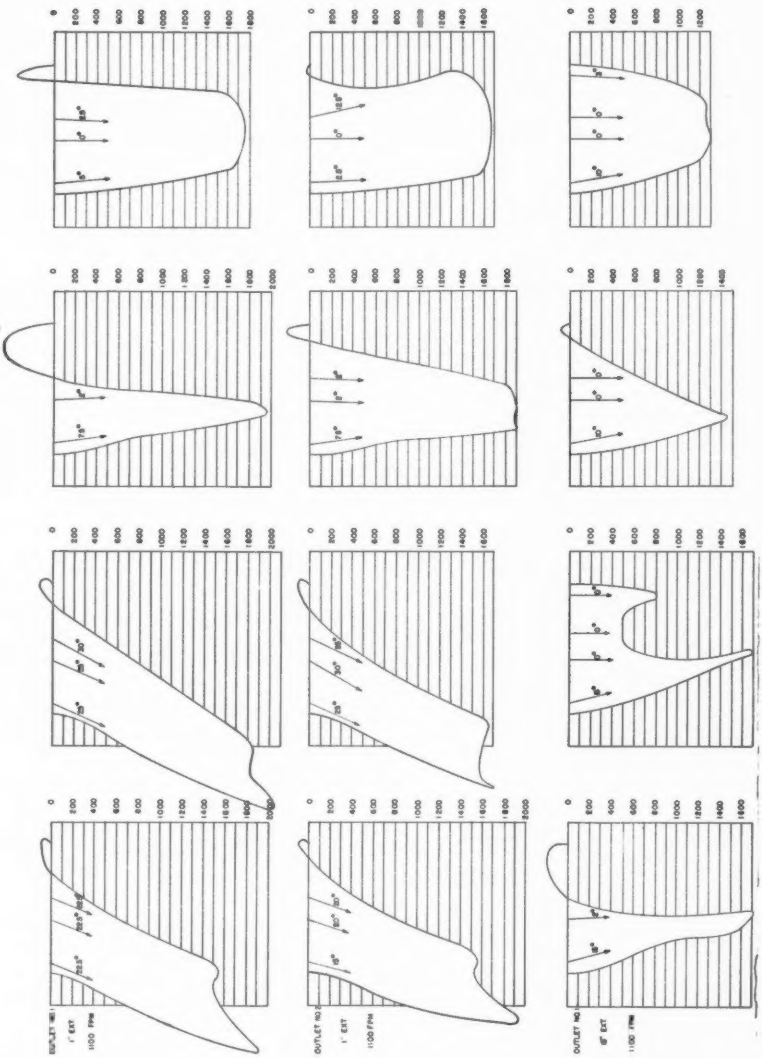


FIG. 5. FLOW PATTERNS FOR 4-IN. x 9-IN. OUTLET


6" X 6" OUTLETS
PLAIN OUTLETS
SINGLE SCOOP OUTLETS
VERTICAL STRIP OUTLETS
CURVED DEFLECTOR OUTLETS
SCOOP
OUTLET NO 1
1" EXT
500 FPM
OUTLET NO 2
1" EXT
500 FPM
OUTLET NO 1
13" EXT
500 FPM
OUTLET NO 2
13" EXT
500 FPM


OUTLET NO 2
1" EXT
1100 FPM

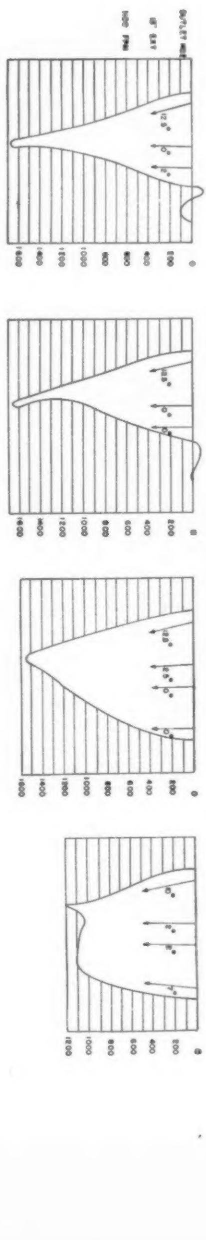


FIG. 6. FLOW PATTERNS FOR 6-IN. x 6-IN. OUTLET

before the outlets and devices of this sort appear to have no particular value. When used on 3 in. x 10 in. outlets, the vertical strip and curved deflector devices eliminated the negative flow area completely and resulted in an air discharge approximately perpendicular to the face of the outlet. Almost the same results were found on the 4 in. x 9 in. and the 6 in. x 6 in. outlets, except that a small area of negative flow remained when these two directive devices were used with a 1 in. extension on the outlets. However, even this small negative area was removed when longer extensions were used.

The flow characteristics of outlet No. 3 are shown in Fig. 7. This outlet was without any directive device during all the tests, but had the same length of extension as on the side outlets in any particular test. The flow from the outlet at the end of the duct was reasonably uniform for each of the three outlet sizes tested. Directive devices on an outlet at the end of a main duct or branch are not necessary. In cases where it is desirable that the air stream be directed, the use of vertical strips set at the proper angles would seem desirable.

Photographs taken in the air flow analyzer for three lengths of extension operated in conjunction with plain, single scoop, vertical strip and curved deflector types of outlets are shown in Fig. 8. The size of the duct used in the analyzer was one-half that of the duct used in the test and the outlet corresponded to 4 in. x 9 in. Fig. 8A shows the very pronounced angle of the air stream from the plain outlet with a 1 in. extension, and Fig. 8B shows that this condition is not corrected appreciably by use of a single scoop inside of the duct. Figs. 8C and 8D show the corrective effect of the vertical strips and curved deflectors.

The large negative flow area with a 5 in. extension is shown in Figs. 8E and 8F. The vertical strips and curved deflectors eliminated the negative flow as shown in Figs. 8G and 8H. It will be noticed from Figs. 8 I and 8 J that an extension of 13 in. on the plain outlets was sufficiently long to eliminate the negative flow area. The curved deflector and vertical strip outlets gave the same results with this length extension as they did with the other lengths.

The amount of negative flow in inches is plotted as a linear distance from the upstream edge in Fig. 9 for various lengths of extensions for the 3 in. x 10 in. side outlets. The results with 4 in. x 9 in. and 6 in. x 6 in. outlets were similar. The plain outlets and the outlets with single scoops showed an increase and then a decrease in the negative flow area as the extension length was increased. This behavior is shown by the flow pictures in Fig. 8. The air envelope on the upstream side of the extension has a pronounced curvature

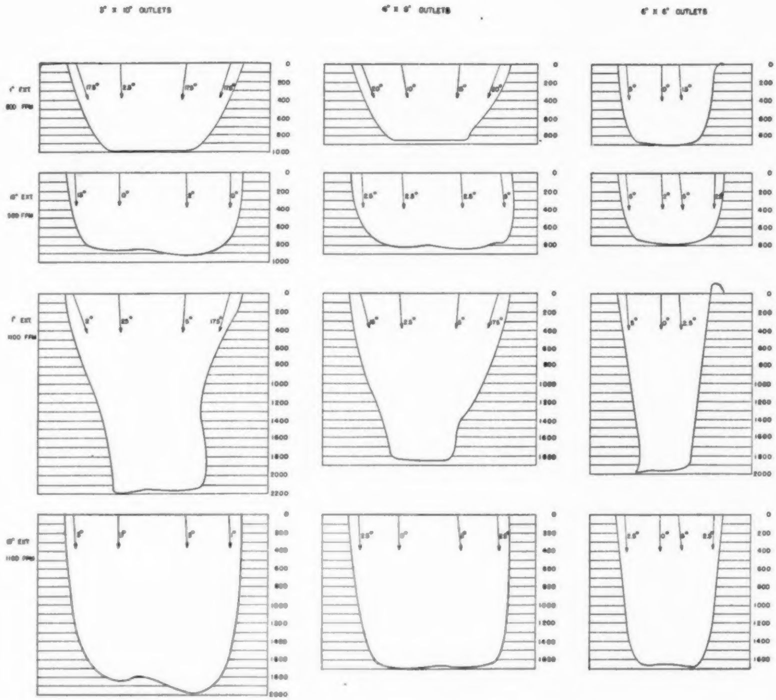


FIG. 7. FLOW PATTERNS FOR OUTLET NO. 3

which results in the complete spread across the face as the extension is increased in length.

In Fig. 10 the angle from the outlet face is plotted against the length of extension in inches for the 3 in. x 10 in. outlets. The angle of flow for the curved deflector and the vertical strip devices placed at the outlet is practically perpendicular to the outlet face for all lengths of extension. However, some further correction takes place in these cases with the use of the longer extension, but not as pronounced as with the plain and single scoop outlets. In the

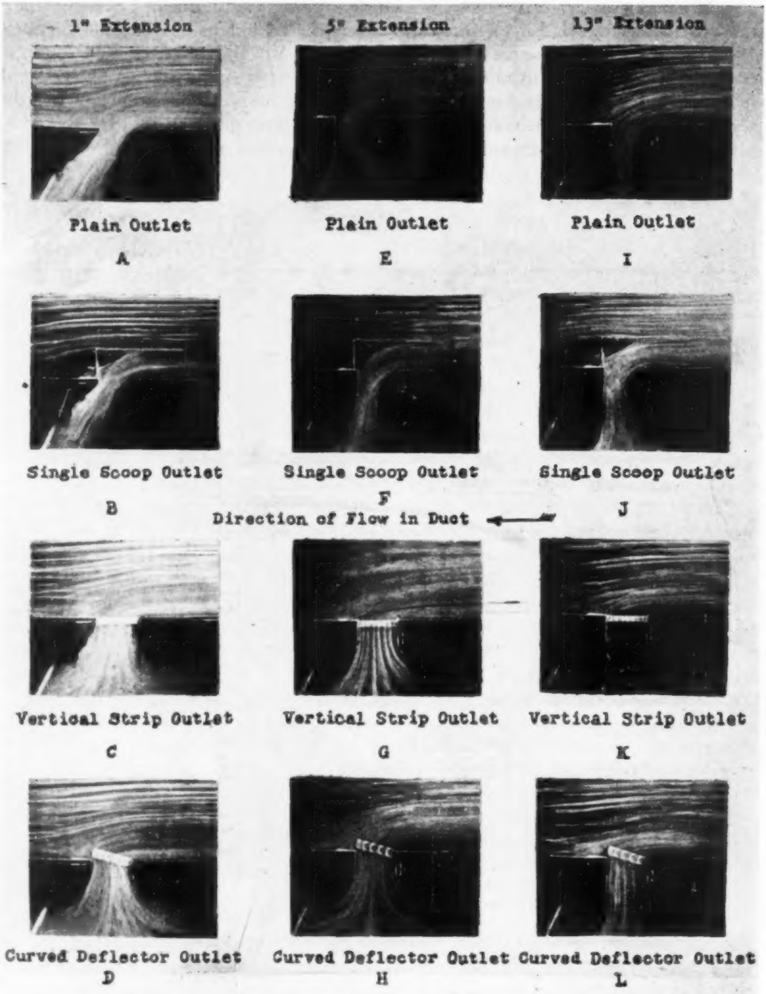


FIG. 8. AIR FLOW PHOTOGRAPHS WITH VARIOUS TYPES OF OUTLETS AND EXTENSION LENGTHS (6-IN. x 20-IN. DUCT AND 4-IN. x 9-IN. OUTLETS)

case of the latter, the increase in the extension length from 1 in. to 10 in. caused the angle of flow for outlet No. 2 to change from 38 deg to perpendicular to the outlet.

The static pressure at point No. 2 is plotted against the duct velocity before outlet No. 1 in Fig. 11. The static pressure at this point is highest when the single scoop device is used before the outlet. This is reduced somewhat as the length of extensions is increased, but still is as high as for the plain outlets without long extensions. The curved deflectors in conjunction with the 13 in. extension gave the lowest static pressure in every case. At a velocity of 200 fpm the change of static pressure was very small, but at 1400 fpm duct velocity the reduction secured by the use of the curved deflectors with 13 in. extension was 50 per cent greater than that obtained with a single scoop.

In Fig. 12 the static pressure at outlet No. 1 is plotted against the length of extension. The length of extension has an appreciable effect on the static pressure in the center of the duct in line with the outlet face. The reduction by the use of extensions is more pronounced with the vertical strips and curved deflectors than with the plain outlet. This reduction in static pressure can be expected to decrease beyond a certain length of extension due to additional frictional resistance in the longer extension.

In Figs. 13, 14, 15, and 16 the variation of static pressure at twelve different points of the system equipped with the 3 in. x 10 in. outlet and various directive devices is plotted for velocities of 500 and 1100 fpm. In each case, except for the single scoop device, definite static pressure regain across the face is indicated. With the single scoop arrangement accurate readings of the static pressures were difficult to obtain across the face of the outlet in the duct. This was due to the single scoop being inserted at this point. However, a regain takes place as is evidenced by the increase of static pressure from point 2 to 7, and to 11 in Fig. 14. The static pressure regain shown in Figs. 13 to 16 is of the same nature as that obtained in transition sections with gradual expansion. In the case of transition sections, the size of the duct is increased with a resultant decrease in velocity. This velocity decrease is accompanied by a conversion of the velocity head to static head with a resultant static pressure increase. In the case of the horizontal duct used in the present investigation, the duct size was held constant but the flow of air from the side outlets caused the reduction in the velocity past the outlet. There was as a result, an increase in the static pressure in the system past each outlet.

CONCLUSIONS

The use of curved deflectors resulted in the best flow characteristics from the standpoint of uniformity of flow, negative area, and static pressure. The vertical strips were not quite as effective as the curved deflectors. The latter have an advantage in that they can be placed at any angle to get the angle of flow desired. The single scoops do not correct the flow pattern over that of the plain outlet, and they do not appear to be desirable in duct installations. Several of these placed parallel to each other at the outlet face would possibly result in improved performance, but they would be less effective and more complicated than vertical strips. The use of volume dampers and grilles was not covered in the present investigation.

ACKNOWLEDGMENTS

The tests on which this paper are based extended over a period of approximately two years. Acknowledgment is due to the following students in Mechan-

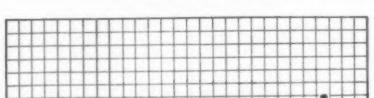


FIG. 9. NEGATIVE FLOW PLOTTED AGAINST LENGTH EXTENSION

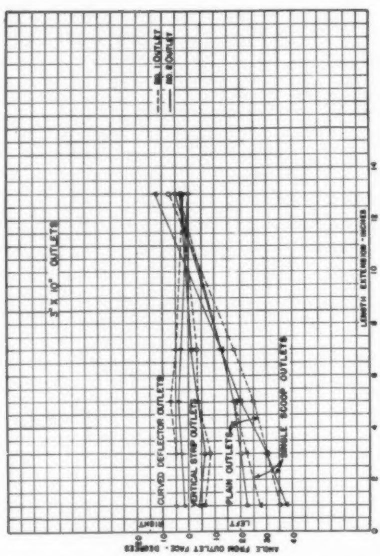


FIG. 10. ANGLE OF FLOW PLOTTED AGAINST LENGTH EXTENSION

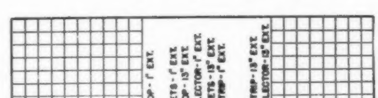


FIG. 11. STATIC PRESSURE AT POINT NO. 2 PLOTTED AGAINST DUCT VELOCITY BEFORE OUTLET NO. 1

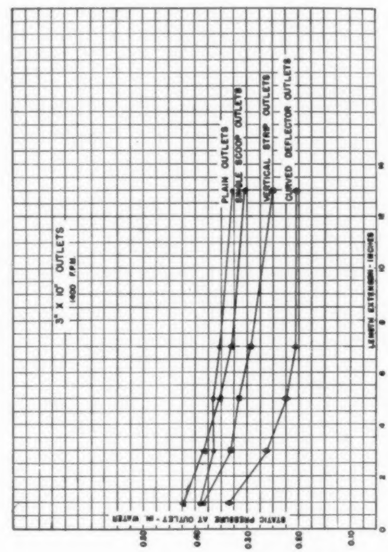


FIG. 12. STATIC PRESSURE AT OUTLET NO. 1 PLOTTED AGAINST LENGTH EXTENSION

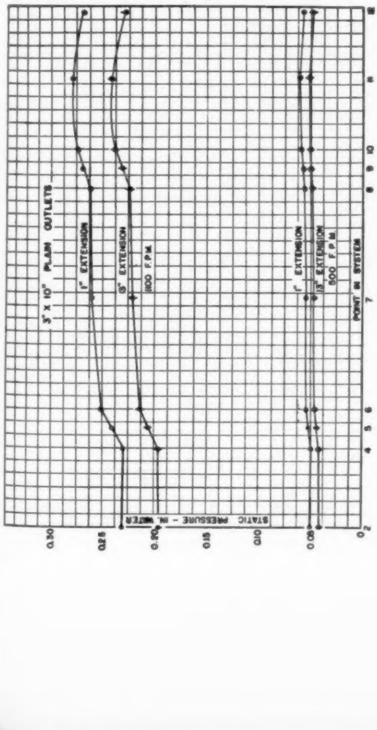


FIG. 13. CHANGE IN STATIC PRESSURE FOR DUCT WITH 3-IN. x 10-IN. PLAIN OUTLETS

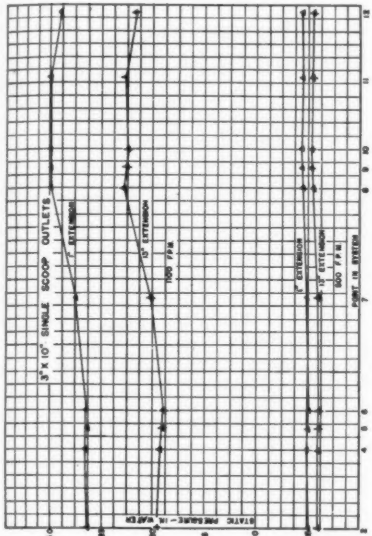


FIG. 14. CHANGE IN STATIC PRESSURE FOR DUCT WITH 3-IN. x 10-IN. SINGLE SCOOP OUTLETS

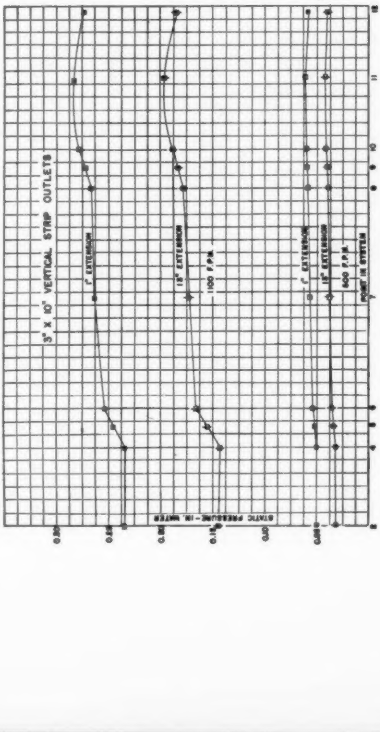


FIG. 15. CHANGE IN STATIC PRESSURE FOR DUCT WITH 3-IN. x 10-IN. VERTICAL STRIP OUTLETS

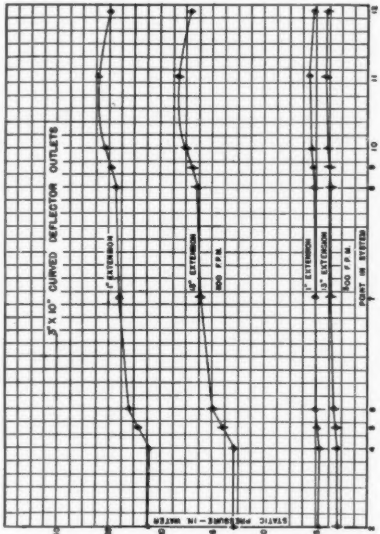


FIG. 16. CHANGE IN STATIC PRESSURE FOR DUCT WITH 3-IN. x 10-IN. CURVED DEFLECTOR OUTLETS

ical Engineering for their valued assistance in the work: A. C. Holmes, C. L. Kaiser, W. H. Rowe, and H. L. Thies.

DISCUSSION

F. H. GEER, Chicago, Ill. (WRITTEN): I would like to know the meaning of the vertical scales in Figs. 4, 5 and 6, and why the straightener was used in the apparatus as indicated in Fig. 2? I feel the results would be more valuable if the test set-up had been just like field conditions.

PROFESSOR NELSON: The 7.55 in. nozzle was installed in the 19 ft length of 18 in. x 18 in. duct leading from the fan to the 6 in. x 20 in. test section in order to measure the volume of air passing through the duct during a test on side outlets. The *egg crate* straightener was inserted up-stream from the nozzle to even out the flow of air from the fan before it entered the nozzle. This was a precautionary measure to insure that a previously determined nozzle coefficient would apply.

A field installation would not have this nozzle but its inclusion in the laboratory set-up is not considered to affect the results since the closest side outlet was down-stream by 15 ft. Installations in the field differ widely one from another so it is necessary to decide on some one laboratory set-up to test. In this case a uniform approach was agreed upon, simulating as nearly as possible side outlets preceded by a long length of straight duct. Had a 90-deg turn immediately preceded the test section the results would have been different—how different only another series of tests could tell. With turning vanes or splitters in the turn the difference in results would be slight. In applying laboratory results the designer or installer must use judgment to cover differences in arrangements used.

The scales to the right of each chart in Figs. 4 to 7 show velocity at the face of the outlet in feet per minute. For instance, in Fig. 6 third column and the fourth graph from the top, the velocity distribution at the face of 6 in. x 6 in. outlet No. 2 equipped with vertical strips is shown for a 13 in. extension and a duct velocity up-stream of the outlets of 500 fpm. The highest velocity at the face was 700 fpm which was found near the downstream edge of the outlet.

1227

No. 1227

ARMY FUEL CONSUMPTION STUDIES OF 1941-42

By L. C. McCabe,* Washington, D. C., S. Konzo,** Urbana, Ill., R. E. Biller,† Washington, D. C.

Introduction

THE Army is one of the largest consumers of fuel in the continental area of the United States. Each camp is equivalent in size and number of buildings to a town or city. The aggregate of hundreds of camps containing thousands of buildings, each of which requires fuel for space heating, cooking, water heating, or process steam presents a fuel demand and a fuel problem of great magnitude. Such phases of the problem as fuel selection, fuel specifications, procurement of funds, fuel inspection, ash disposal, and heating plant maintenance require most careful planning and execution. One of the major problems has been the proper allotment of fuel for these purposes. Too small a fuel allotment may result in sudden overload demands for fuel in winter, too large an allotment may result in carelessness in use, loss by spontaneous combustion, overload of limited storage space, and unjustified inroads on the industrial and domestic supply. The problem is complicated because camps are located in areas with all gradations between year 'round heating and short seasonal demands; almost every available type of fuel is used; the personnel changes continually; every type of heating equipment is used; and buildings ranging from permanent hospital structures to tents are heated.

In the winter of 1941-42, the Office of the Chief of Engineers of the United States Army made a study[1] of the problem. The University of Illinois, Engineering Experiment Station, provided consulting service and the U. S. Weather Bureau supplied the necessary weather information. This paper presents the general nature of attack of the problem and the method of determining fuel requirements recently adopted by the Army.

Limitations of Former Method of Fuel Allotment

Prior to the present building program of the United States Army, the determination of the total quantity of fuel required for heating buildings was made by the application of Equation (1).[2]

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* Major, U. S. Army, Office of Chief of Engrs., Construction Div., Repairs and Utilities Br.
** Consultant, Special Research Associate Professor, University of Illinois, Member *of* A.S.H.V.E.
† Associate Engineer, U. S. Army, Office of Chief of Engrs., Construction Div., Repairs and Utilities Br.

[1] This study was authorized by Col. G. F. Lewis, Chief of Repairs and Utilities Br., Office of Chief of Engrs., and was directed by Maj. L. C. McCabe, Chief of Heating and Refrigeration Section of that office. Acknowledgment is made to W. A. Hepburn, Capt. J. B. Cordiner, and J. J. Bowe, Office of Chief of Engrs. who participated in the study. The Weather Bureau studies which will be reported in separate publications were under the direction of J. P. Kohler and H. C. Thom. Professors S. Konzo and A. P. Kratz, Department of Mechanical Engineering, University of Illinois, served as specialist consultants.

[2] Army Regulations No. 30—1620, Quartermaster Corps, issued June 10, 1932.

Presented at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1943.

The term *Hours of Operation at Total Capacity* was based on average temperatures as shown by U. S. Weather Bureau records and predetermined factors compiled by the Quartermaster General. Specific details as to the method used in determining the *Hours of Operation* are not available, but by reconstruction of existing data it is probable that the term was derived from Equation (2).

With reference to the term *Standard Fuel*, the following quotation from Army Regulations Number 30-1620 is of interest.

"The standard fuel referred to is anthracite coal having a content of 12,500 Btu per pound and not more than 10 per cent of ash nor more than 10 per cent of volatile matter. This coal has been taken as standard for the reason of its uniformity of quality, but this standard is not to be construed as indicating that it is to be used in preference to any other fuel."

$$\left\{\begin{matrix}\text{Lb Standard Fuel}\\ \text{per Building}\\ \text{per Season}\end{matrix}\right\} = (\text{Factor}_1)\,(\text{Factor}_2)\left\{\begin{matrix}\text{Hr of Operation}\\ \text{at Total Capacity}\end{matrix}\right\} \quad . \quad . \quad . \qquad (1)$$

in which,

a. For steam heat, Factor_1 = 0.04 lb standard fuel per hour per square foot of direct radiation.

Factor_2 = square feet of direct radiation or its equivalent, including surface of uncovered pipe used for heating the building, exclusive of risers.

b. For hot water heat,

Factor_1 = 0.0272

Factor_2 = square feet of direct radiation.

c. For warm-air furnaces, stoves and fireplaces,

Factor_1 = 5 lb of standard fuel per hour per square foot of grate surface.

Factor_2 = square feet of grate surface.

$$\left\{\begin{matrix}\text{Hours of Operation}\\ \text{At Total Capacity}\\ \text{per Season}\end{matrix}\right\} = \frac{24 \times N \times (70 - t_a)}{(70 \times t_d)} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

in which,

24 = hours per day

N = number of days in heating season

t_a = average outdoor temperatu reduring heating season, in degrees Fahrenheit

t_d = outdoor temperature assumed for design of plant, in degrees Fahrenheit

For purposes of converting the allowance, expressed in standard fuel as shown in Equation (1) to the fuel actually in use, Equation (3) was used:

$$\text{Lb Fuel Actually Allowed} = \frac{\text{Lb Standard Fuel}}{\text{Equivalent factor}} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

The extensive table of equivalent factors given in the same publication for all types of fuel shows that for solid fuels the equivalent factor became smaller as the ash content and volatile matter content increased.

The method of determining fuel allowances based on the use of the preceding equations was reasonably satisfactory, particularly after actual operating experience indicated the modifications necessary to meet local conditions.

During 1940-41 a critical analysis of the prevailing method of determining fuel allowance indicated that the method was deficient when applied to newer camps composed largely of temporary type buildings, and that a thorough revision of the method to conform to present-day commercial practice was desirable. Some of the deficiencies in the prevailing method may be enumerated as follows:

1. The numerical values for $Factor_1$ in Equation (1) were determined from a limited number of tests[3] on permanent type buildings located in and near Washington, D. C. and probably would not be applicable to the temporary type structures that constitute a large proportion of the new army building program.

2. The use of the $Factor_2$ involving the square feet of installed radiation was equivalent to basing the fuel allowance on the calculated heat loss. For a given type of building, the amount of installed radiation would be different for each design temperature zone, so that for purposes of checking, it was considered more desirable to obtain a constant that would be uniformly applicable over the country and which would be independent of the design outdoor temperature.

3. The allowance of 5 lb of coal per hour per square foot of grate surface for warm-air furnaces and stoves was not considered applicable to present-day equipment

FIG. 1. U. S. ARMY FUEL CONSUMPTION STUDY POSTS OBSERVED

or to the temporary type buildings. In addition, the wide range of furnace sizes used over the country to heat even the same type of building would make the process of checking fuel allowances an exceedingly laborious task.

4. The numerical values for *hours of operation at total capacity* depended not only upon the mean temperature during the heating season and the length of the heating season, both of which could be obtained from the Weather Bureau, but also upon the design outdoor temperature, which quite frequently is chosen arbitrarily by the engineer.

5. The extensive tables of equivalent factors for coal incorporate the effects of ash, volatile matter, and size of coal, but neglect both the calorific value and fixed carbon content. The evaluation of the relative effectiveness of coals has been considered as a separate problem and results obtained will be presented separately.

6. The fuel study demonstrated that with certain adjustments in quantities provided by Army Regulations 30—1620, the per capita basis was the most satisfactory basis for computing fuel allowances for laundry, water heating and cooking.

OBJECTIVES OF FIELD STUDIES

In view of the limitations inherent in the methods of determining total fuel requirements outlined in Army Regulations 30-1620, the Office of the Chief of

[3] *Fuel Allowances*—A Thesis Presented for the Degree of Mechanical Engineer by James Henry Platt, B.M.E., the Ohio State University, 1924.

Engineers authorized a field study of fuel consumption at actual field installations. The following objectives were considered in planning the test program:

1. A method utilizing the degree-days for each locality was to be used for space heating.
2. A constant designated as the *space heat fuel allowance,* or the pounds of standard fuel per 1000 sq ft of floor area per degree-day, was to be determined by test for the various types of buildings.
3. A simplified procedure was to be established for calculating the fuel requirements for heating all buildings.
4. Additional data were to be gathered upon which to establish revised allowances for water heating, cooking, and laundry purposes on the per capita basis.

METHOD OF PROCEDURE

Selection of Buildings for Study: Located geographically as indicated on map (Fig. 1), 10 different camps were selected for the field studies. These camps represent wide variations in weather conditions and utilize the three general types of fuel such as coal, gas and oil.

Since the buildings in any given post may be many hundreds in number, and the types of buildings may exceed 100, an intensive fuel survey of each building at each post was prohibitive in time and expense. Instead, the total number of buildings to be studied at each post was limited to from 30 to 40, and by judicious selection a representative sample of the various types of buildings and fuel usage was obtained. The barracks-type structure constitutes the largest proportion of the total number of buildings at any post, in some cases com-

TABLE 1—TYPICAL LIST OF BUILDINGS UNDER OBSERVATION

Fort DEVENS **State** MASS.

Type of Building	Sq Ft Floor Area	No. of Bldgs. Observed
TEMPORARY BUILDINGS		
Barracks: 63-Man	4,720	3
Mess: 118 Officer	1,980	1
Mess: 210-Man	2,664	2
Administration: A-5	1,144	1
Administration: A-22	3,200	1
Exchange: E-3	3,663	1
Officers' Quarters: OQ-40	7,670	2
Recreation: RB-1	3,663	1
Motor Repair Shop: SP-2	3,108	1
Infirmary: I-2	2,056	2
Company Storehouse and Administration: SA-2	1,296	1
Theater: TH-3	10,724	1
Storehouse: SH-13	9,190	1
PERMANENT BUILDINGS		
Officers' Private Quarters: OQ	2,044	2
Non-Commissioned Officers' Private Quarters: MCOQ	858	2
Enlisted Man's Barracks: EMB	66,000	1

prising over 50 per cent of the total floor area at the post. As shown in the typical list given in Table 1, from 3 to 6 barracks were placed under observation at each project, and an average fuel consumption was obtained for this predominant type of structure. As a result of the sampling procedure, approximately 90 to 97 per cent of the total floor area at any post can be considered as included in the study. Fig. 2 presents several of the types of buildings studied showing the general construction.

FIG. 2. TYPES OF BUILDINGS STUDIED SHOWING GENERAL CONSTRUCTION

Observation of Fuel Consumption: An engineer, located at each of the 10 posts, recorded weekly fuel consumptions of from 30 to 40 separate buildings during the heating season of 1941-1942. In all cases the fuel supply to the buildings under study was separately recorded, and precautions were taken to insure that the fuel assigned to a given building was not diverted to some other building or use. A typical weekly summary sheet for all the buildings is shown in Fig. 4.

In these studies no attempts were made to vary or improve the normal routine and handling of the heating plants and the buildings. The data obtained may be considered as typical of actual operation in the field.

Factors Affecting Fuel Consumption for Space Heating: The weekly fuel consumption for any given building and heating plant will be dependent upon the following climatic variables: (1) Temperature difference between indoors and outdoors. (2) Wind effects. (3) Sun effects. (4) Moisture effects.

Of these four variables, the difference in temperature between indoors and outdoors is the predominant factor,[4] so that if the weekly fuel consumption is plotted on a curve against the temperature difference, or the equivalent degree-

FUEL CONSUMPTION CHART

POST- DEVENS   COAL- BUCKWHEAT

POST BLDG. NO.- 1614   FUEL USE- SPACE HEAT

REMARKS- 2 OLD BOXES, EACH HOLDS 65#/IN.

Form 327

DATE	COAL ADDED	TOTAL COAL IN BIN	COAL VOL.	DATE	COAL VOL.	ESTIMATED CONSUMPTION	LBS. LEFT OVER
OCT. 1		3270	49½"	OCT. 8	36½"	840	2300
OCT. 14	3850			OCT. 15	65"	1860	4290
				OCT. 22	50½"	950	3280
OCT. 27	2870			OCT. 29	67"	1800	4350
				NOV. 5	42"	1625	2725
NOV. 7	2025			NOV. 12	43"	1950	2800
NOV. 18	2100			NOV. 19	43"	2100	2800
NOV. 25	4020			NOV. 26	67"	2470	4350
				DEC. 3	26"	2660	1690
DEC. 4	3160			DEC. 10	30"	2900	1950
DEC. 11	4130			DEC. 17	35"	3805	2275
DEC. 18	3180			DEC. 24	26½"	3735	1720
DEC. 24	3340			DEC. 31	28"	3240	1820

FUEL DELIVERIES TO BUILDING

DATE	SEPT. 22	SEPT. 22					
AMOUNT-LBS.	1900	1900					
DEPTH AFTER	29"+	29"					
" BEFORE	0	0					
INCHES	29	29					
LBS/IN.	65.5	65.5					

FIG. 3. FUEL CONSUMPTION CHART

[4] In University of Illinois Engineering Experiment Station Bulletins Nos. 189 and 246, data are presented showing the effects of wind and sun on fuel consumption in Research Residence. In extremely high winds fuel consumption is greater than normal. There are no adequate means, however, of obtaining an over-all index of weather that would properly evaluate the effects of the four variables. Furthermore, it is questionable whether a single index of such a nature would be generally applicable to all types of building structures. As a result of this difficulty, the only practical method has been to plot fuel consumption against the predominating factor, which is temperature difference, and then to extend the period of fuel observation over a long period of time, so that periods of high wind, low wind, high sunshine, low sunshine, rain, snow, and sleet are all included. A curve drawn through the average of points secured in this manner will represent average weather conditions for the locality. This procedure requires a test period that may extend over several months. Experience has indicated the danger of making conclusions based upon test periods that are less than a month in duration. In the final analysis of test data secured from all 10 posts, the average wind and sun effects were tabulated and scrutinized, but no consistent results could be observed which might indicate that a special correction factor for either factor was required. An examination of weather data for the United States indicates that with the exception of a few localities, the average wind velocity in winter months does not exceed about 12 mph. Later experience may indicate that an empirical correction should be applied to fuel allotments for these few exceptions.

days for the week, a correlation can be expected. Typical curves showing data for barracks fired with coal are shown in Fig. 5.

It may be observed from Fig. 5 that the curves tend to converge to the zero point. An examination of the data for all types of buildings and from all 10 posts indicated that heating was required whenever the mean outdoor temperature ranged from about 63 to 67 F, so that a base temperature of 65 F for the determination of the degree-day values appears to be adequate for this type of service. The only exception was in the case of hospital buildings and clinics, in which a base temperature of 70 F seemed to be more appropriate. Rather than introduce a new base temperature for degree-day calculations,

REPORT NO. 13

POST Fort Devens

DATE Dec. 19, 1942

SUPR. Leo D. [illegible]

OBSERV PERIOD	BLDG. NO.	BLDG. TYPE	NO. OF OCC.	TOTAL AREA	FUEL BTU VALUE: SPACE HEAT	HOT WATER	UTILITY	FUEL CONSUMPTION LBS.: SPACE HEAT	STD. FUEL	HOT WATER	STD. FUEL	UTILITY	STD. FUEL	TOTAL STD. FUEL	DEG. DAYS	SPACE HEAT FACTOR	LBS.S.F. PER UTIL PER M.H.	TOTAL FUEL FACTOR
12, 1-8	T-1616	E-3	--	3663	12,950	--	--	2520	2610	--	--	--	--	2610	186	3.8	--	3.9
12, 1-8	T-1637	63M	47	4720	12,950	13,350	--	1980	2050	450	480	--	--	2530	186	2.3	1.5	2.9
12, 1-9	T-1640	63M	37	4720	12,950	13,350	--	1685	1748	600	641	--	--	2386	186	2.0	2.8	2.7
12, 1-8	T-1641	63M	27	4720	12,950	13,350	--	1760	1820	640	685	--	--	2505	186	2.1	3.1	2.9
12, 1-8	T-1643	A-3	--	1144	12,950	--	--	490	507	--	--	--	--	507	186	2.4	--	2.4
12, 1-8	Hosp.	Area	1017	280780	1	4,1	00	2	6 3 ,	3 6 5	--2	7 4,0	00	274,000	186	--	--	5.3
12, 2-9	T-1682	OQ-40	30	7670	12,950	13,350	--	3960	4100	370	395	--	--	4495	178	3.0	1.9	3.3
12, 2-9	T-1651	OQ-40	27	7670	12,950	13,350	--	4180	4340	375	400	--	--	4740	178	3.3	2.1	3.6
12, 2-9	T-1650	OM-116	55	1980	13,380	13,350	14,000	492	510	308	329	1980	2220	3059	178	1.5	6.6	8.7
12, 2-9	T-1633	210M	180	2664	12,950	14,000		1265	1310	3390		-	3800	5110	178	2.8	3.0	10.8
12, 2-9	T-1639	210M	200	2664	13,350	14,000		1270	1318	2520		-	2820	4138	178	2.8	2.0	8.9
12, 3-10	T-1613	A-22	34	3800	12,950	--	--	2467	2560	--	--	--	--	2560	180	4.5	--	4.5
12, 3-10	T-1614	EE-1	--	3663	12,950	--	--	2900	3000	--	--	--	--	3000	180	4.5	--	4.6
11,26;12,10	P-40	OQ	3	2044	12,950	528/cu.ft.		3800	3320	4490	--	189		3509	374	4.3	4.5	4.6
11,26;12,10	P-41	OQ	3	2044	12,950	528/cu.ft.		3420	3540	3950	--	167		3707	374	4.8	4.0	4.9
12, 4-11	T-1617	I-2	20	1980	12,950	--	--	2956	3060	--	--	--	--	3060	182	8.0	--	8.0
12, 4-11	P-112A	Barr	130	16500	--	14100	588	--	--	1500	1695	18100	765	2460	182	--	2.6	--
11,29;12-12	P-112	Bar.	300	66000	14,100	--	--	71,940	81000	--	--	--	--	81,000	429	2.9	--	--
12, 5-11	T-1854	TH-3	--	10724	14,100	--	--	Space Heating begun Dec.	5				--	No Reading Taken				
12, 5-12	T-1827	SH-13	2	436	12,950	--	--	157	160	--	--	--	--	160	223	1.7	--	1.7
12, 5-12	T-1830	SA-2	6	1188	12,950	--	--	940	975	--	--	--	--	975	223	3.7	--	3.7
11,14;12,12	P-110	NCOQ	3	868	12,950	528/cu.ft.		2370	2460	3800	--	148		2608	700	4.1	1.8	4.4
11,14;12,12	P-108	NCOQ	2	868	12,950	528/cu.ft.		2850	2950	4800	--	178		3128	700	4.9	3.2	5.2

Form [illegible]

FIG. 4. DEGREE-DAY STUDY FIELD DATA

merely to take care of a relatively small part of the total heating requirements, a uniform base temperature of 65 F, which is the commercially recognized value, was accepted and fuel requirements for hospital purposes determined on a partially different basis. The latter will be discussed more in detail presently.

It is apparent from Fig. 5 that as the severity of the weather increased, the weekly coal consumption also increased. Furthermore, in general, the consumption did not increase proportionally with increases in the degree-days for the week. Relatively less coal was used per degree-day in severe weather than in mild weather. This experience, which is different from that frequently found in tests of field installations of domestic and commercial buildings, is considered in more detail in the following sections:

Determination of Space Heat Factor: The basic data shown in Fig. 5 are not in a form suitable for later application, since in applying the data for the purposes of estimating fuel consumption it would be necessary to obtain the degree-days per week for each week in a given locality, and then to obtain the fuel consumption for each week from the figure. A more generalized factor was required that would facilitate the application of data.

One obvious generalization would be to determine a factor, F, which would have units of *pounds of standard fuel per degree-day per Btu per hour heat loss,* and which would be used in Equation (4). The difficulty with this equation is that the last term, involving the calculated heat loss of the building, is not a constant for the same type of building. The calculation of heat losses

$$\left\{\begin{array}{c}\text{Lb Standard Fuel}\\ \text{per Building}\\ \text{per Season}\end{array}\right\} = (F)\left\{\begin{array}{c}\text{Degree-}\\ \text{days per}\\ \text{Season}\end{array}\right\}\left\{\begin{array}{c}\text{Design Heat Loss}\\ \text{of Building in}\\ \text{Btu per hour}\end{array}\right\} \quad . \quad . \quad . \quad . \quad (4)$$

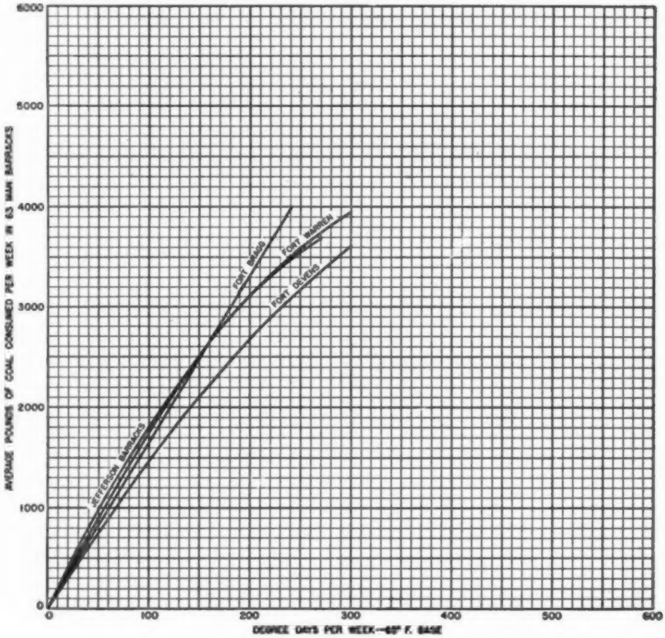


FIG. 5. COAL CONSUMPTION FOR BARRACKS TYPE BUILDING

from a building is not immune from individual interpretations, nor are outdoor design temperatures positively determined for each and every part of the country. Hence, in spite of the fact that a single factor, F, might be obtained which would be applicable to more than one type of building on the post, the obvious difficulties in obtaining and checking the design heat losses for even a single type of building for several hundred posts eliminated from consideration this obvious approach.

A barracks building of standard design, whether located in Alaska or Florida, may have widely varying design heat losses, but does have one feature of constancy, and that is the physical size. Hence, for the purposes of the Army, a factor based upon some physical dimension of the building was most acceptable.

This factor, which has been termed the *Space Heat Fuel Allowance*, or *SHFA*, is in units of *Pounds of standard fuel per* 1,000 *sq ft of floor area per degree-day*, and is used in Equation (5):

$$\begin{Bmatrix}\text{Lb Standard Fuel}\\ \text{per Building}\\ \text{per Season}\end{Bmatrix} = (SHFA)\begin{Bmatrix}\text{Floor Area}\\ \text{in thousands}\\ \text{of sq ft}\end{Bmatrix}\begin{Bmatrix}\text{Degree-}\\ \text{days per}\\ \text{Season}\end{Bmatrix} \quad . \quad . \quad . \quad . \qquad (5)$$

In this case, the values for floor area are readily available in the Army records and are constant for a given type of Army standardized construction, and degree-day data for each post are available from U. S. Weather Bureau computations for all posts of the Army. Knowing the actual amount of fuel consumed at all test buildings, the floor area of each, and the number of degree-

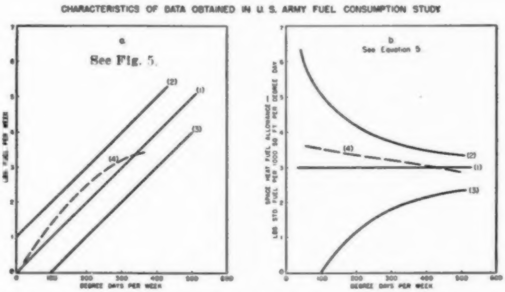


FIG. 6. GENERALIZED DIAGRAMS SHOWING CHARACTERISTIC TRENDS OF SPACE HEAT FACTOR

Case 1. Fuel consumption proportional to degree-days.
Case 2. Fuel required when mean outdoor temperature exceeds 65 F.
Case 3. Fuel required when mean outdoor temperature is lower than 65 F.
Case 4. Fuel consumption not proportional to degree-days.

days during the period of observation, *SHFA* were calculated and used in the final analysis of the data.

Characteristics of Space Heat Fuel Allowance: In Fig. 6*a* are shown generalized diagrams plotted on the same bases as those in Fig. 5. Four separate cases are shown, as follows:

Case 1. Fuel consumption is a straight-line function of degree-days, and is exactly proportional.

Case 2. Fuel is required when the mean outdoor temperature is greater than 65 F and is a straight-line function of degree-days.

Case 3. Fuel is required only when the mean outdoor temperature is less than 65 F and is a straight-line function of degree-days.

Case 4. Fuel consumption is not proportional to the degree-days. The curves shown in Fig. 5 are similar to this case, as were the curves representing the data for other types of buildings.

If the units in the vertical ordinate of Fig. 6*a* are converted to units of *SHFA*, the generalized relationships shown in Fig. 6*b* are obtained. *Case 1* is the only case which would provide a constant value over the entire range of

weather conditions. Actually, nearly all of the data obtained in these studies were similar in character to *Case 4*. That is, for any given post, the *SHFA* increased as the weather became milder, and decreased as the weather became

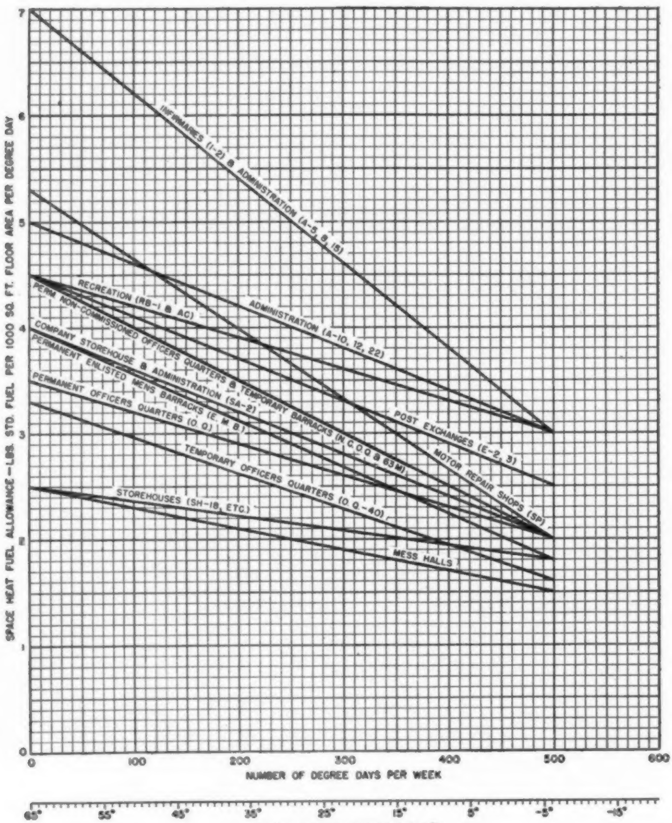


FIG. 7. SUMMARY OF WEEKLY SPACE HEAT DATA FOR INDICATED TYPES OF BUILDINGS

more severe. Furthermore, a given type of building located in the South, where the degree-days were small, as compared with the same type of building located in a colder climate, used relatively more fuel than the ratio of total seasonal degree-days would indicate. This could be accounted for on the basis that any one or all of the following conditions existed:

1. That the efficiency of combustion increased as the heating demand increased. This supposition does not seem tenable in view of the fact that for coal-fired equipment the efficiency usually decreases with an increase in combustion rate.

2. That either due to differences in clothing or differences in acclimatization, a higher indoor temperature is required in early fall and late spring than in the middle of winter. Records of air temperatures in the buildings did not indicate that noticeably higher indoor temperatures were being maintained in milder weather than in severe weather. As a matter of fact, from the standpoint of equal comfort, experience in domestic heating practice has indicated that in severe weather a higher setting of the room thermostat is necessary to counteract the influence of cold walls and cooler temperatures in the living zone.

3. That the heat loss from the buildings was relatively greater per degree difference in temperature from indoors to outdoors for mild weather than for severe weather. This supposition seems to be the most likely explanation of any that have been offered. The prevailing practice is to allow considerable open window ventilation in the building, both night and day. As the weather becomes more severe it is probable that the windows are not opened as widely and that as consequence the interchange of indoor and outdoor air decreases.

If Curves 2 and 3 in Fig. 6*a* represent the upper and lower limits of the range of data, for which Curve 1 is the average, the derived Space Heat Factors will show very wide deviations from a mean value for small numbers of degree-days per week and will show smaller deviations from the mean value for severe weather. This is illustrated in Fig. 6*b* in which the curves for Cases 2 and 3 converge towards the right. A study of the actual data showed that the tendency was characteristic of all the data.

Application of Space Heating Fuel Allowance

Application to Different Temperature Zones: For a given type of building the data from all of the posts were plotted on a single sheet and average curves were drawn through the hundreds of points. The final summary of data for several types of buildings is shown in Fig. 7. This summary represents the end results of one of the most comprehensive field surveys of fuel requirements ever attempted.

It is apparent from Fig. 7 that a constant value for the *SHFA* cannot be used for two buildings of the same type located in widely divergent climatic conditions. If, however, the mean outdoor temperature during the heating season is known, then the corresponding weekly degree-days can be evaluated, and the values for *SHFA* can be read directly from Fig. 7. It has been proposed that all calculations should be based on *SHFA* for a mean outdoor temperature of 45 F, and that corrections should be made for other temperatures depending upon the slope of the line representing the *SHFA*.

For Army purposes, this procedure has been modified somewhat. Based on a correlation of degree-days and the average temperature, the value of the correction factor was made dependent upon the average number of degree-days occurring per season.

Weather Bureau Data: As an integral part of this project the U. S. Weather Bureau supplied the following data for the principal weather stations in the United States adjacent to the posts: (1) Station. (2) Degree-days per year (65 deg base), mean value. (3) Degree-days per year (65 deg base), based on 75 per cent frequency. (4) Degree-days per month for each month, corre-

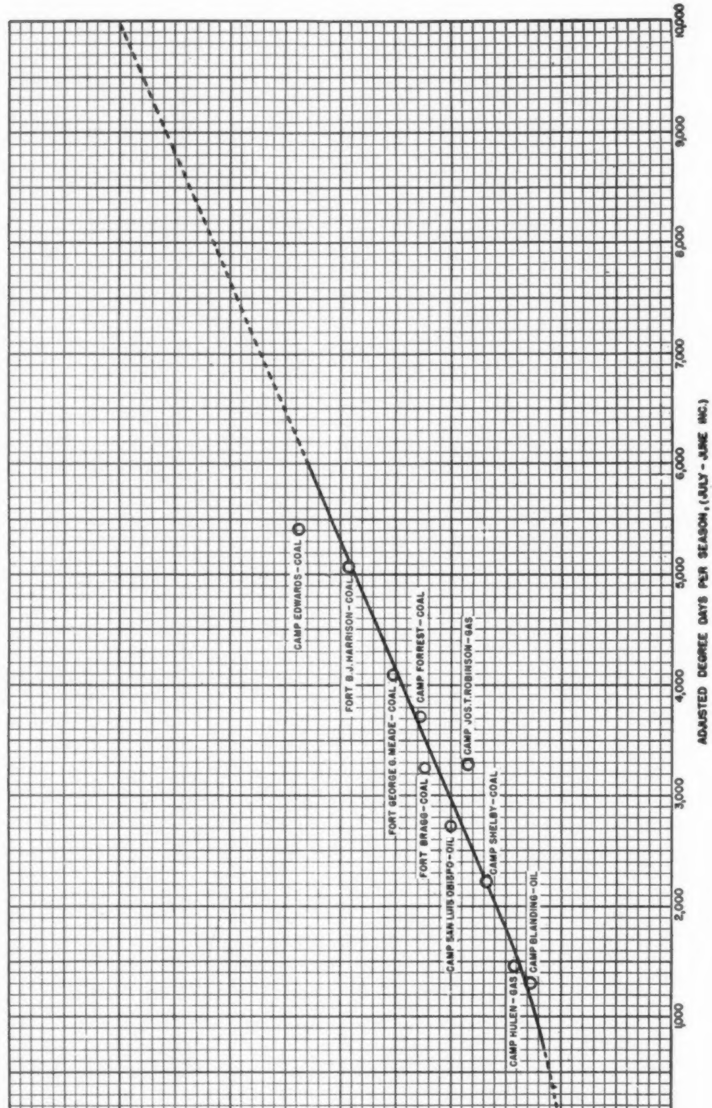


Fig. 8. Hospital Consumption Data

sponding to Item 3. (5) Per cent of yearly degree-day value represented by each monthly value. (6) Starting date of heating season; that is, when mean daily temperature reaches 65 F. (7) Ending date of heating season; that is, when mean daily temperature rises to 65 F. (8) Length of heating season. (9) Mean daily temperature during heating season. (10) Average wind veloc-

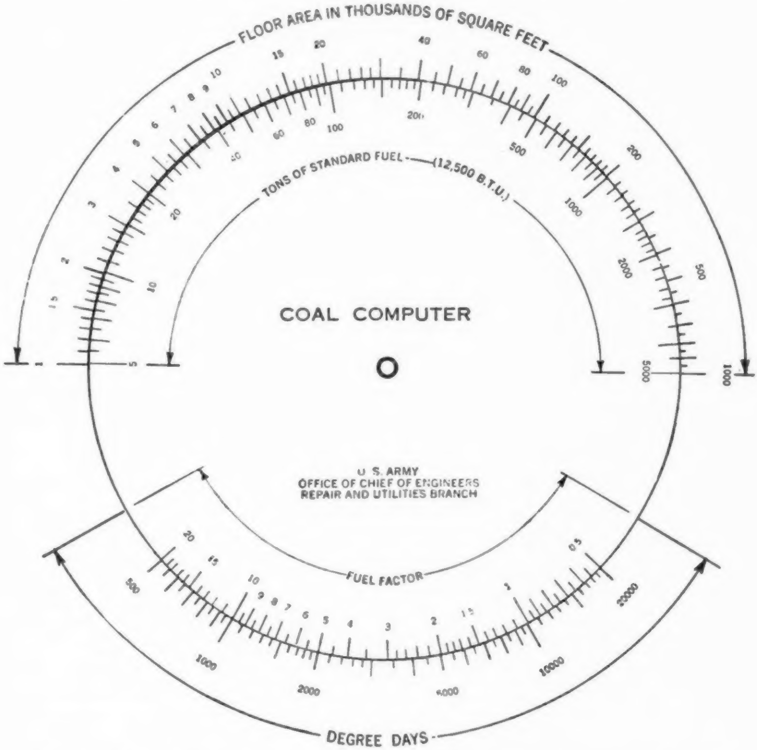


FIG. 9. COAL COMPUTER DESIGNED FOR DETERMINING U.S. ARMY POST FUEL REQUIREMENTS FOR SPACE HEATING

ity during heating season and prevailing direction. (11) Index of sunshine hours during heating season.

Quite extensive tables and charts are currently available[5] that present weather data for the United States. Up to the present time, however, the U. S. Weather Bureau has not officially issued comprehensive tables or maps that cover the preceding items. The official report of the Weather Bureau is to be published at a later date.

[5] *Degree-Day Handbook* and *Air Conditioning Engineers' Atlas* by The Industrial Press.

TABLE 2—BUILDING GROUPS

No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7
TEMPORARY BUILDINGS						
Enlisted man's barracks: 63 M 74 M 45 M 25 M Recreation buildings: RB-1, 2 & 4 ORBL-3 Post exchange: E-1, 2 & 3 CPX-1 Guard houses: GH-1 & 2 UGH-12, 24 & 36 CGH-50	Officers' Quarters with and without mess: OQ-5, 7, 11, 17, 40, BOQ-32, 40, 44 OQM-40 BOQM-28, 36	Officers' and enlisted man's messes: OM-118 EM-72, 112, 152, 170,172, 210, 228, 248, 250, 300 Storehouses: SH-18, 19, 20, 21 (partially heated storehouses not considered) Sheds: MS-2	Administration buildings: A-4 to A-22 ADM-1 IBA-1 Infirmaries and clinics (outside of cantonment hospital area): I-1 to I-5 DC-1 & 2 Service clubs: SC-1, 2, 3 & 4 Schools: SCS-1 SB-12 Trainer buildings: L-7 SH-9 (Mod.)	Motor repair and utility shops: SP-1, 2, 6, 8, 9, 10, 11, 12, 13, 14 MRS-1 & 2 Mechanized shops: AFS-1 BES-1 & 2 TRS-1, 2 & 3 All theater of operation type construction regardless of building type	Storehouse and company administration: SA-1 & 2 Storehouse recreation and company administration: RSA-1, 2 & 3 Company maintenance and arms room: CMS-1	All tentage and hutment construction.
PERMANENT BUILDINGS						
NCO private quarters	Officers' private quarters; administration or headquarters buildings			Hangars: DH-1 & 2 OBH-1 & 2 Post exchanges, theaters, chapels, and any miscellaneous heated permanent buildings	Enlisted man's barracks with or without mess Hospital	

Cantonment Type Station Hospitals

The buildings comprising an Army station hospital differ in one respect in that they are heated centrally from the hospital boiler house.[6] Hot water and some steam for mess purposes are provided from this source also. Because of these conditions, it is necessary to estimate fuel requirements on a different basis, namely, tons of standard fuel per 1000 sq ft of floor area per year.

Table 3—Space Heat Fuel Allowance

Building Group No.	Allowance—Lb Std. Fuel per M Sq Ft Floor Area per Degree-Day
1	3.8
2	3.0
3	2.3
4	5.1
5	4.4
6	3.4
7	7.0

Yearly fuel consumption figures were gathered from station hospitals distributed geographically throughout the continental United States. Knowing the floor area and degree-days for the season, the fuel required per 1000 sq ft floor area per year was calculated for each post. These values were plotted against degree-days per year, resulting in Fig. 8. Table 7 of the appended directive gives a tabular summary of fuel allotment for this service. Station hospitals utilizing coal, gas and oil were included in this survey.

Water Heating, Cooking and Laundry

The fuel required for water heating, cooking and laundry purposes has been determined on the per capita basis, results of which are presented in Tables 8 and 9 of the appended directive. Data for water heating and cooking were collected from the barracks and mess halls, respectively. The per capita fuel requirements for cooking and water heating vary little with the number of men accommodated. Fuel required for laundry purposes was calculated from fuel consumption and post population records. (Table 9.)

Table 4—Climate Factor

Degree-Day Range per Year	Factor
0–3,500	1.08
3,500–7,500	1.00
7,500–up	0.92

Table 5—Fuel Burning Factor

Fuel	Factor
Coal	1.00
Gas	0.88
Oil	0.93

[6] An Army cantonment type station hospital consists of a number of small, separate one-story buildings (a 1000-bed hospital of this type will normally consist of 75 to 80 buildings). All are interconnected by means of enclosed walkways. In these walkways are located the mains of the steam distribution system.

GENERAL NOTES

As might be expected from the wide spread of locations and fuels encountered in Army practice, considerable variations in fuel consumptions from an average value were obtained even for the same type of building. The empirical approach discussed in this paper, in which the average of hundreds of plotted points was obtained, does provide a starting point from which a reasonable fuel estimate can be made. The use of the mean seasonal degree-days as a basis of obtaining the normal fuel consumption, likewise enables the Office of the Chief of Engineers to provide for an increased fuel allotment when the actual heating season is more severe than normal. If experience indicates that the fuel consumption for any given post is consistently greater than the average allotment for that post, the combustion engineers will at least be able to concentrate their attention on that post to find the reason for the variance.

At one post, in which the fuel used was petroleum coke, the fuel consumption was considerably greater than when coal was used. Petroleum coke is extremely responsive to changes in draft and it is questionable whether such a fuel can be efficiently burned in Army service unless extra precautions are taken to train the firemen in the proper control of the drafts and in the proper method of handling the fuel.

TABLE 6—FUEL CONVERSION FACTORS

COAL		GAS		OIL		
Btu per Lb	Tons Actual Fuel per Ton Std. Fuel	Btu per Cu Ft	M Cu Ft Gas per Ton Std. Fuel	A P I Grav.	Btu per Gallon	Gals Oil per Ton Std. Fuel
15,000	0.83	450	55.5	10	154,600	167
14,750	0.85	475	52.6	12	153,300	163
14,500	0.86	500	50.0	14	152,000	164
14,250	0.88	525	47.6	16	150,700	166
14,000	0.89	550	45.5	18	149,400	167
13,750	0.91	575	43.5	20	148,100	169
13,500	0.93	600	41.7	22	146,800	170
13,250	0.94	625	40.0	24	145,600	172
13,000	0.96	650	38.5	26	144,300	173
12,750	0.98	675	37.0	28	143,100	175
12,500	1.00	800	31.3	30	141,800	176
12,250	1.02	825	30.3	32	140,600	178
12,000	1.04	850	29.4	34	139,400	179
11,750	1.06	875	28.6	36	138,200	181
11,500	1.09	900	27.8	38	137,000	183
11,250	1.11	925	27.0	40	135,800	184
11,000	1.13	950	26.3	42	134,700	186
10,750	1.16	975	25.6	44	133,500	187
10,500	1.19	1,000	25.0	46	132,400	189
10,250	1.22	1,025	24.4	48	131,200	191
10,000	1.25	1,050	23.8	50	130,100	192
9,750	1.28	1,075	23.2	...	...	...
9,500	1.32	1,100	22.7	...	...	...
9,250	1.35	1,125	22.2	...	...	...
9,000	1.39	1,150	21.7	...	...	...
...	...	1,175	21.3	...	...	...
...	...	1,200	20.8	...	...	...

Two regions in the United States have weather characteristics that differ appreciably from those in the rest of the country. These regions are the San Francisco Bay area and the Puget Sound area in Washington. In both regions a mean daily outdoor temperature less than 65 F may be obtained in every month of the year. Hence, with a hand-fired plant there may be a tendency to keep the fires going and to burn some fuel even when no heating demand exists, merely to save the labor of relighting the fires every time the weather turns cool. For such regions the fuel consumption may be greater than the number of degree-days would indicate is necessary. Experience may indicate the necessity of providing for a small factor of safety for the coal consumed during such holdover periods.

TABLE 7—CANTONMENT TYPE HOSPITAL ALLOWANCE

Degree-Days per Year	Allowance—Tons Std. Fuel per M Sq Ft Floor Area per Year
0	10
500	11
1,000	12
2,000	16
3,000	20
4,000	25
5,000	29
6,000	33
7,000	38
8,000	42
9,000	46
10,000	50

TABLE 8—HOT WATER HEATING AND COOKING ALLOWANCE

Use	Allowance—Tons Std. Fuel per Man per Year		
	Coal	Gas	Oil
Hot water heating	0.30	0.25	0.30
Cooking.........	0.45	0.15	0.25

TABLE 9—LAUNDRY ALLOWANCE

Fuel Used	Allowance—Tons Std. Fuel per Man per Year
Coal	0.14
Gas	0.10
Oil	0.11

The following directive for computation of fuel allowances (degree-day data deleted), which was released by the Adjutant General on October 9, 1942, illustrates the application of the fuel consumption study to the determination of fuel requirements for Army posts, irrespective of magnitude and location.

COMPUTATION OF FUEL ALLOWANCES[7]

This study demonstrates that the total fuel requirements of a post are divided functionally into the following items:

A. Space heating for all buildings except the cantonment type hospital area. (See Fig. 9 for Coal Computer Slide Rule.)

B. Total fuel required for the cantonment hospital.

C. Total fuel required for hot water heating and cooking except the cantonment hospital area.

D. Fuel for laundry purposes.

E. Fuel purchased for resale.

F. Other miscellaneous uses.

[7] Inclosure No. 1 to Memorandum SPX 463, (10-2-42) SPEGC-MP-R-M, October 8, 1942.

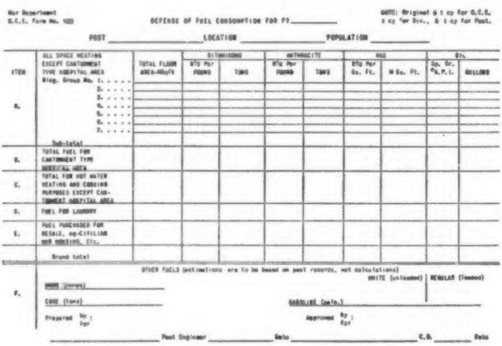
GENERAL DISCUSSION

1. *Building groups:* All buildings have been classified into 7 groups according to their fuel requirements for space heating. Table 2 shows the building types for each group.

2. *Fuel allowance factor:* The fuel allowance factor for space heating per 1,000 sq ft of floor area per degree-day was determined for each group of buildings from field data. These allowances are given in Table 3.

3. *Degree-days:* The degree-day is a unit used to express heating requirements when the daily average temperature falls below 65 F. If the average temperature of a 24-hour period is 45 F, the heating requirements of this period are equivalent to 20 degree-days. Daily degree-day values are added to obtain yearly totals. The number of degree-days assigned to calculate heating requirements in certain localities are tentative, pending determination of exact values by the U. S. Weather Bureau. These tentative values include a factor of safety of such magnitude that they will rarely be exceeded. Table 1 contains a list of posts and stations together with an adjusted number of degree-days for each location.

4. *Climate factor:* Fuel consumption data indicate that as the outdoor temperature becomes milder, more fuel is required to heat a given unit of space per degree-day.


War Department
O.C.E. Form No. 423

DEFENSE OF FUEL CONSUMPTION FOR FY.________

NOTE: Original & 1 cy for O.C.E. 1 cy for Div., & 1 cy for Post.

POST ________ LOCATION ________ POPULATION ________

ITEM		TOTAL FLOOR AREA	BITUMINOUS		ANTHRACITE		GAS		OIL	
			BTU Per POUND	TONS	BTU Per POUND	TONS	BTU Per Cu. Ft.	M Cu. Ft.	Sp. Gr. °A.P.I.	GALLONS
A.	ALL SPACE HEATING EXCEPT CANTONMENT TYPE HOSPITAL AREA Bldg. Group No. 1.									
	2.									
	3.									
	4.									
	5.									
	6.									
	7.									
	Sub-total									
B.	TOTAL FUEL FOR CANTONMENT TYPE HOSPITAL AREA									
C.	TOTAL FOR HOT WATER HEATING AND COOKING PURPOSES EXCEPT CANTONMENT HOSPITAL AREA									
D.	FUEL FOR LAUNDRY									
E.	FUEL PURCHASED FOR RESALE, eg-CIVILIAN AND HOUSING, Etc.									
	Grand total									

F. OTHER FUELS (estimations are to be based on past records, not calculations)

WOOD (cords) ________

COKE (tons) ________

GASOLINE (gals.) ________ WHITE (unleaded) | REGULAR (leaded)

Prepared By: For

Approved By: For

________ Post Engineer ________ Date ________ C.O. ________ Date

FIG. 10. O.C.E. FORM NO. 423

A climate factor has been introduced which allows more fuel per unit floor area per degree-day in a warmer locality. The value of this factor varies with the total number of degree-days per year as indicated in Table 4.

5. *Fuel burning factor:* This factor compensates for differences in the fuel burning efficiency of coal, gas, and oil as indicated in Table 5.

6. *Fuel Conversion factor:* This factor is used to convert standard fuel quantities to quantities of fuel as burned at the post. See Table 6.

7. *Cantonment type hospital allowance:* Table 7 contains the fuel allowance for cantonment type hospitals per 1,000 sq ft of floor area per year. If the adjusted degree-day value falls between values shown in the table, factor values will be interpolated to obtain the correct allowance.

8. *Hot water and cooking allowance:* Table 8 contains separate fuel allowances for hot water heating and cooking on a per capita basis, when either coal, gas, or oil is used.

9. *Laundry allowance:* The fuel allowance for laundry purposes is on the per capita basis. Table 9 contains the allowances for coal, gas or oil.

10. *Form No. 423* (Fig. 10) : This form will be executed annually in quadruplicate. The original and one copy will be forwarded by January 1 to the Chief of Engineers, Attention of Repairs and Utilities Branch. One copy will be retained by the division engineer and one by the post engineer. This form will also accompany all supple-

mental requests for fuel due to additional housing capacity. Only the additional construction or other items on which the estimate of additional fuel is based need be listed on the supplemental form.

DIRECTIONS FOR COMPLETING FORM NO. 423 (FIG. 10)

1. Fill in spaces for fiscal year, post, location, and population. The latter includes all officers and enlisted men.
2. Group all buildings at the post in the respective classifications as indicated in Table 2, and compute the total floor area of each group in M (thousands) square feet. Insert these values in total floor area column.
3. Sample calculation for items A to F required on Form No. 423 (Fig. 10). Assume Post X is located in a 6,000 degree-day area and the post population is 40,000 men. Buildings in group 1 have a total floor area of 500,000 sq ft. Bituminous egg coal containing 13,500 Btu per lb is used for space heating, cooking, and hot water heating. The cantonment type hospital has a floor area of 180,000 sq ft. Bituminous coal having 13,000 Btu per lb is used at the hospital and laundry boiler houses. (The Btu value of the coal will be assumed to be the average *as received* value of the coal used for that purpose during the preceding year.)

ITEM A. Space heating. (Similar calculation to be made for each building group.)

Formula: (fuel allowance from Table 3) × (floor area, M sq ft) × (degree-days per year from Table 1) × (climate factor from Table 4) × (fuel burning factor from Table 5) × (fuel conversion factor from Table 6) divided by 2,000-tons, MCF gas, or gallons oil.

$$3.8 \times 500 \times 6{,}000 \times 1.00 \times 1.00 \times 0.93 \div 2{,}000 = 5{,}300 \text{ tons}$$

Enter 5,300 tons under the bituminous heading on Form No. 423 opposite building group No. 1.

ITEM B. Total fuel for cantonment type hospital.

Formula: (fuel allowance from Table 7) × (floor area, M sq ft) × (fuel burning factor from Table 5) × (fuel conversion factor from Table 6) = tons, MCF gas, or gallons oil.

$$33 \times 180 \times 1.00 \times 0.96 = 5{,}700 \text{ tons}$$

Enter 5,700 tons on line B in bituminous coal column.

ITEM C. Total fuel required for hot water heating and cooking except cantonment hospital area.

Formula (separate calculations for hot water and cooking are required where two fuels are used): (fuel allowance from Table 8) × (post population) × (fuel conversion factor from Table 6) = tons, MCF gas, or gallons oil.

$$0.75 \times 40{,}000 \times 0.93 = 27{,}900 \text{ tons bituminous,}$$
$$\text{for hot water heating and cooking purposes.}$$

Enter 27,900 tons in bituminous column for Item C.

ITEM D. Fuel for laundry purposes.

Formula: (fuel allowance from Table 9) × (post population) × (fuel conversion factor from Table 6) = tons, MCF gas, or gallons oil.

$$0.14 \times 40{,}000 \times 0.96 = 5{,}375 \text{ tons bituminous}$$

Enter 5,375 tons in bituminous column for Item D.

ITEM E. Fuel for resale may be computed as in Item A knowing the type of building to be heated.

ITEM F. Other fuels are entered as indicated.

DISCUSSION

G. H. TUTTLE, Detroit, Mich (WRITTEN): An analysis of the information presented in this paper would seem to indicate that a better and more complete method of heat regulation would be very desirable in all army camp buildings.

Theoretically, the heat consumption of any building, regardless of occupants, location, or type of construction, is proportional to the indoor-outdoor temperature difference. This being the case, the heat consumption of buildings for any period should be proportional to degree-days so long as the same indoor temperature is maintained for all outdoor temperatures. Fig. 6, which presents pounds of coal per week compared with weekly degree-days for barracks in four camps, indicates that in one case, at least, this relationship is true. In the case of Fort Bragg, the pounds of coal consumed per degree-day is approximately 16.5 for all weather conditions from the warmest to the coldest. This indicates that the operation is consistent. However, in Fig. 10, where the total fuel allowance is presented, Fort Bragg is above the average of other camps so it must be consistently high. In terms of operation, this would seem to indicate that perhaps fairly constant indoor temperatures were maintained but that either the indoor temperatures were maintained higher than necessary or for longer hours.

If it is possible to maintain a constant heat consumption per degree-day for all outdoor temperatures in one camp, it seems probable that the same could be done for other camps, regardless of geographical location and average outdoor temperatures.

Further evidence to the fact that apparently indoor temperatures are maintained higher than necessary, or at least the average heat loss is higher than necessary, is indicated from Tables 3 and 4 where *Fuel Allowance* figures are presented. In Table 3, the average fuel allowance per degree-day, excluding group No. 7, is 3.7 lb per 1000 sq ft of floor area. If this ratio is applied to a five-room house with 1000 sq ft of floor area in a geographical location where the annual degree-days are 6500, the expected coal consumption for a normal year would be in excess of 12 tons per year. General practice would indicate that a home owner would have to be very extravagant to consume this amount of coal.

The total fuel allowance in tons per year for hospital-type buildings is presented in Table 4. For 6500 degree-days, the total fuel allowance would be 35.5 tons of coal per 1000 sq ft. By comparison to this figure, a hospital served with district steam where accurate metered steam consumption data are available, used 81,000 lb of steam annually per 1000 sq ft. In terms of coal burned at 60 per cent efficiency, this consumption would be 5.4 tons per year, or approximately one-sixth of the fuel allowance value given in Table 4.

The civilian population of the United States, particularly those living in the east and midwest, are quite conscious at the present time of fuel rationing. With all the information available in this report, the opportunity of applying the OPA fuel oil formula is more than the average civilian can resist. Assuming the conditions in the previous example, and applying it to No. 1 oil with 138,000 Btu per gallon, the OPA formula would reduce the oil consumption from 2200 gal per 1000 sq ft per year to 1161, or a reduction of 47 per cent.

The authors are certainly to be commended on their method of presenting the voluminous amount of data that were collected for the report. It is just one more example of the fact that war is a big business and that the supply problem, perhaps, is not as spectacular as other phases of the war effort, but is just as important.

W. A. DANIELSON, Memphis, Tenn.: As a matter of historical interest on this question of fuel allowances, this work was started back about 1915 by the Quartermaster Corps of the United States Army. It is very interesting to the Society because the tests that were made at that time at Pittsburgh in the Laboratory were a factor later in the establishment of the Society's Laboratory there. A great number of tests were run to determine the amount of fuel burned per square foot

of radiation. That was finally determined quite accurately for the class of fireman that was brought in from all over the Army. The square feet of radiation was used because that was the best measure of the heat loss of the particular building.

From that we went next to the only method that we had in those days of determining the effect of climate. We divided the United States up into nine zones where there were about equal average temperatures through the heating season. We had no degree-days in those days. We had to figure fuel allowances for every conceivable type of building and not just the barracks.

This scheme is limited only to a type of building and a certain class of construction. We started to figure or to transform the previous system over to the degree-day basis about five or six years ago, but there was not much incentive to change, because our previous method had been fairly accurate.

N. D. ADAMS, Rochester, Minn.: This paper is very interesting from the point that it brings out a factor we have known for years about space heating, particularly as to greenhouse construction, which is true if you apply the same principle to all types of buildings. All types of buildings are more or less porous, and, as the degree-days increase, we get a frosting or a closing up of the porous type of construction.

I want to bring out at this point that there is something the authors have missed in this paper; the difference in humidity will change the frosting and as the increase in humidity goes up the frosting increases and closes up faster. Therefore, you will have a greater decrease in the amount of heating required at the greater number of degree-days.

GENERAL DANIELSON: I would like to differ with Mr. Adams just a bit. I want to bring into the record that this frosting up that he mentioned was due to the doctors requiring the windows to be exceedingly wide open.

L. E. SEELEY, New Haven, Conn.: There is a publication by the U. S. Housing Authority making a study of its properties in various parts of the United States. As I remember the figures expressed in terms of Btu per room per degree-day, the properties in the South, where the degree-days were low, ran about 9,000; the properties in the North ran about 4,900, showing quite a difference per degree-day, which substantiates this paper.

But of even greater interest is the following: after trying to effect economies they discovered that they could reduce the heat required in Southern properties from 9,000 to 7,000, which is a good saving; but in the Northern property they could only go from 4,900 down to 4,600, which is a very small saving. I think that comparisons with other types of heating should not be made. The character of the buildings, *i.e.* barracks, the type of use and the maximum use of window ventilation affords no basis for judging the OPA fuel rationing program.

R. L. DAVISON, New York, N. Y.: We have just studied a group of apartment houses in connection with fuel rationing, and found the oil consumption over the last three years more closely correlated with 70 F used as a base for degree-day, than 65 F base. We also found that heat was controlled in the more favorable apartments, having southern exposure or close to the boilers, by opening the windows. I think that explains some of the reasons why you have such wide variations in these curves. There is a widespread practice of controlling temperatures in apartment houses by opening windows, but it is not so true of individual houses where you pay for your own heat.

F. E. GIESECKE, College Station, Tex.: The difference between southern and northern localities, of which Professor Seeley spoke, may result partly from the fact that there are many days in the South when heat is needed in buildings but when, under the 65 degree method, no degree-days are recorded. Heat requirement calculations for southern localities should be based on the number of degree-hours instead of on the number of degree-days in the heating season.

J. W. MILLER, Lansing, Mich.: In Lansing one of the oil companies delivers fuel oil by the degree-day method, and they found they could do a better job of delivering oil using 70 as the basis for degree-day calculation rather than 65.

V. H. WILSON, Nashville, Tenn.: When the housing project idea was really started in the southern states they eliminated space heating entirely. Camp Forrest was also constructed with the idea that it was to be based on 10 deg above 0. Incidentally, we have temperatures in our locality that go far below that figure.

The other item is that our low temperature days down in the south do not last for periods of three, four and five days at those temperatures. It goes down overnight from maybe 35 or 40 deg above to 0 one night and then the next day at noon it would be back up to 50 deg. So, naturally, the peak in the locality requires tremendous amounts of heat in hotels and public buildings due to that great fluctuation. You will have to go back to degree-hours, and not degree-days.

JAMES HOLT, Cambridge, Mass.: Were all the barracks buildings referred to by the authors designed on the basis of the mobilization building plans with the same type and thickness of insulation, throughout the buildings?

MR. BILLER: It is true as stated by Mr. Tuttle that a better and more complete method of reducing fuel requirements would be desirable in all Army camp construction. It is to this end that the Army posts have established firing schools for the training of firing personnel. Also, experienced civilian personnel have been hired in supervisory capacity to insure that the heating equipment is properly maintained.

However, to insure that no more heat is required for a given building than indicated by its design heat loss would mean design conditions would have to be maintained continually. For instance, only the design amount of crack area would be permitted. The openings of doors and windows would be strictly regulated. Any number of property owners will testify that tenants not responsible for their heat bills are inclined to require more fuel than that indicated as necessary according to the design heat loss of the building. It is doubtful if civillians, who in general do not adhere to the strictest principles of heat conservation under comparable conditions would expect Army officials to enforce such regulation upon the soldier personnel.

The following information may serve to explain why the fuel allowance for an Army cantonment type hospital is considerably in excess of the fuel requirements of that hospital served by district steam, referred to by Mr. Tuttle. I presume that this civilian hospital is of the construction common to civilian life today, *viz.*, relatively few number of buildings of brick and stone construction. In contrast to this, an Army cantonment type station hospital of 1,000 bed capacity will normally consist of 75 to 80 small separate one-story buildings of wooden construction. All are interconnected by means of inclosed walkways. In these walkways are located the mains of the steam distribution system which distribute steam for heating to all buildings of the area from a central boiler plant. Because of this extensive steam distribution system, one can easily appreciate the excessive line loss that must occur in the system.

This information is not offered as an apology for the fuel requirements of the station hospital, but it is presented in order to explain the difference in construction and lay-out between the Army cantonment type station hospital and the hospital common to civilian life.

All temporary barracks studied were of the mobilization type uniformly constructed according to design. In general, all temporary buildings under observation were of the "700" series construction. At this time when the fuel study was initiated this construction was prevalent.

Relative to the time when this study was begun some confusion apparently exists. The work discussed today was planned in the early part of 1941 and conducted for one year without interruption including the winter season of 1941-42.

The work referred to as having been initiated in 1915 is separate and distinct from that considered today, although the object of both endeavors was the same—to establish a basis for making Army fuel allowances. It was considered necessary to

establish a new basis for fuel allotment not because the former was unscientific or inaccurate but because it was not readily applicable to the new type of construction.

The essential difference between the two methods may be summarized as follows: With the old method the fuel allotted for space heating was dependent upon the calculated heat loss of the building, a quantity that might vary according to the particular engineer making the calculations. With the recent method the quantity of fuel allotted is not dependent upon any calculation but rather has been derived from mass fuel consumption tests conducted at ten Posts located throughout the United States. Inasmuch as the quantities of fuel consumed were actually measured, the resulting fuel consumption data are independent of theoretical differences and personal opinions.

The data on the barracks were presented as an example because this type of structure is most common on the Army Post. Over 50 per cent of the total floor area of the average Post consists of barracks. We studied a total of approximately thirty types of structures at all Posts.

Regarding the basis for calculating degree-days the data obtained from all the Posts for practically all types of buildings indicated that the most satisfactory point lay between 63 F and 67 F. Therefore, 65 was selected. If the data had indicated that 70 F would give results that provided better correlation, we would have chosen this value inasmuch as our prime purpose was to evolve data that would best serve as a basis for the most representative fuel allowances.

Several comments were made on the fact that the data indicate less fuel is required per degree-day as the number of degree-days occurring per week increase or, in other words, as the weather becomes colder. Based on personal observations of the soldier's living habits and general reports from others, I am convinced that reduction in open window ventilation as the outside temperature becomes more severe is chiefly responsible for the smaller amount of fuel required per degree-day for heating purposes in colder weather.

Another factor which undoubtedly contributes to this tendency to some extent is the fact that degree-days as determined by the conventional methods do not take into account all factors which contribute to the need for space heating. For example, in sections of the country where clouds are fewer and terrestrial radiation is considerable the night-time temperature may easily drop to a level where heating is required yet the daytime temperature may soar to such heights as to wipe out the effect of the low night temperature when a calculation is made to determine the degree days occurring in the period.

Data from Posts located in the south substantiate these statements. Degree-hours were calculated in an effort to increase the degree-days occurring in a given period and thereby attempt to equalize the fuel required per degree-day in mild and cold weather. However, in general the difference was not of sufficient magnitude to compensate for the extra effort involved.

Also, there is the physiology difference between inhabitants of warm and cold climates to consider. Although this factor is impossible to evaluate, yet it cannot be disputed that the average person of a warm climate will require external heat for bodily comfort at a higher temperature than will an individual accustomed to more severe temperatures.

S. KONZO: The comparison of army fuel consumption with normal residential service will always be more favorable to the domestic application, particularly if the buildings are assumed to be about the same. As a matter of fact, from the standpoint of a heat barrier, the army temporary structures are really shells that serve as good windbreaks, but are far from being heat-tight structures. The 700 series and 800 series buildings are mostly siding on building paper, with no insulation and no inner wall. The design heat losses for a 63-man barracks type building runs 346,000 Btu per hour for a 70 F difference and 447,000 Btu per hour for a 90 F difference. Since the floor area is 4,720 sq ft it may be noted that the Btu per square foot

runs about 84 for an 80 deg design temperature difference. This value is much higher than any ordinary residence structure. A large part of this loss can be attributed to the liberal ventilation allowance provided by the Army design engineers.

I have attempted to calculate the overall efficiency of fuel consumption based upon the average curves shown in Fig. 5, and have found that the overall efficiency runs from 75 to 85 per cent. Admittedly these calculations assume the validity of the design heat loss calculations, but they do give an inkling of the effectiveness of utilization. The high fuel consumptions are largely the result of the rather porous construction of the buildings, rather than poorly operated or controlled heating plants.

The hospital buildings are distinctly not comparable in size or construction with the ordinary hospital. In addition the large number of small structures are connected with covered walks, under which the steam pipes run and which are heated indirectly from the heat loss from the pipes. The consumptions, therefore, include not only the building consumption but the piping loss, which must be considerable.

Mr. Tuttle's comments with relation to the straight-line relation of fuel consumption to degree-days are most pertinent and is in agreement with our experience in the two research residences on the campus. Contrary to expectations, however, in practically all cases for the army buildings the fuel consumption per degree-day decreased as the weather became colder. We have devoted considerable space discussing this very point, and Fig. 6 was drawn to explain the nature of this deviation from our expectations.

The only rational explanation that we can give to the fact that the consumption per degree-day decreased as the weather got colder, is given in explanation 3 on page 85. The Army engineers have pointed out that open window ventilation is widely used, regardless of temperature control, and that such window openings are not under control. It is the opinion of the field engineers that as the weather gets colder the windows are not opened so widely. If this is the case the actual heat losses in mild weather will be larger in proportion to the temperature difference than in cold weather.

One factor that was not brought out in the condensed paper was the large variability in fuel consumptions for two identical buildings side by side. This was true even with gas-fired equipment under thermostatic control. The variations were so large that it was exceedingly difficult to base any conclusions on one building alone. In general the average values given in Fig. 7 of the paper represent data from a large number of buildings of the same type. Any single building may deviate quite widely from the average for that type of building. The data presented merely summarizes the statistical average, with the hope that the values apply to large groups of buildings, but not necessarily to any one specific building.

The point might be very logically raised as to why the Army does not attempt to build better buildings from the standpoint of heat conservation. The only answer to that question is that the design group for the Army engineers has the job of building hundreds of camps consisting of thousands of buildings, within a budgetary allowance not established by them. Admittedly, most of the buildings are not being built with the idea of having them last 20 years. As far as the fuel division of the Army is concerned the question is only of theoretical interest. Their job is to heat and maintain what is furnished to them. The fuel division might conceivably prove that a better structure would save fuel. However, they do not have the authority to revise building specifications to effect such conservation. Inasmuch as these buildings are for relatively short term use, the economics of investing larger sums for insulation, weather stripping, sidewalls, and false ceilings simply does not prove itself.

I hope that these few comments will bring out the distinction between normal civilian usage and Army usage. They differ considerably, as we realize only too well. The final criterion, and the only one during the emergency, is whether or not the camps can be built at a rate fast enough to take care of the situation and within a reasonable budget. As far as I can judge the Army engineers have met that criterion.

1228

No. 1228

PERFORMANCE CHARACTERISTICS OF A COAL-FIRED SPACE HEATER

By R. C. Cross,* Chicago, Ill.

The 1940 census of the United States,[1] made by the U. S. Department of Commerce, shows that 58 per cent of approximately 34 million dwelling units in the United States are heated by stoves or means other than central heating plants. These stoves and space heaters may be divided into two general classes: radiant heaters and circulating heaters. The latter type has found favor in more recent years because some heating effect is obtained in rooms adjacent to the room in which the heater is located.

Circulating heaters are available with gravity and forced circulation, and special types have been developed for the various domestic fuels—gas, oil, and solid fuels. It is estimated that approximately 10 per cent of total space heater installations are gravity circulating heaters fired with solid fuels. Fig. 1 shows a typical heater of this classification.

There is a definite lack of published information and specific data in regard to the performance characteristics of this type of heater, and the test program that forms the basis of this discussion was carried out in an effort to supply some of this information.

General Program

Comparative information was obtained by determining the output of tests with anthracite, coke, lignite, coking bituminous coal, and free-burning bituminous coal. A smoke recorder also was used to obtain a log of the smoke produced with each fuel.

General Summary

The following general conclusions are derived from the test results:

1. A wide range of solid fuels can be burned successfully in the conventional space heater.
2. A considerable difference in rate of burning, available firing period, and output was found with the different fuels. These factors are dependent upon the reactivity of the fuel, all other factors remaining constant.
3. When firing high volatile fuels an excessive amount of smoke is produced by this type heater. This is an inherent disadvantage of the overfeed principle of burning.

Test Set-Up

The 18 in. heater used in these tests has 9.03 sq ft of heating surface and a grate area of 0.80 sq ft. Ratio of heating surface to grate area, 11.26 to 1.

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* Senior Combustion Engineer, Sears, Roebuck and Co. Member of ASHVE.

[1] Sixteenth Census of the United States, U. S. Department of Commerce.

Presented at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1943.

The heater was set on scales so that the rate of burning could be determined directly by weight loss. The instruments necessary for measuring drafts and temperatures, flue gas collection and analysis, and photoelectric smoke recorder were connected in accordance with standard practice. Fig. 2 shows the general arrangement of this equipment.

TEST PROCEDURE

The data taken included the weight of fuel charged, weight of ash and refuse, ashpit draft, draft at flue gas outlet, flue gas composition, temperature of air

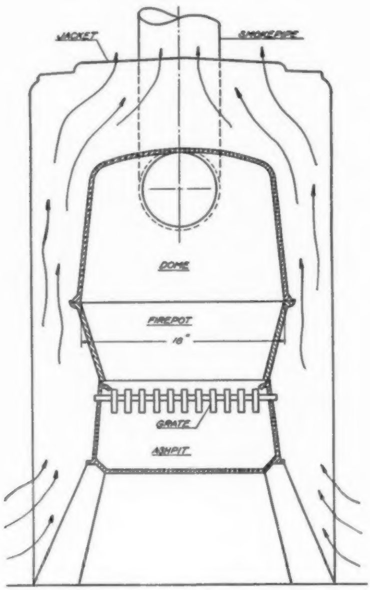

FIG. 1. TYPICAL CIRCULATING HEATER

for combustion, flue gas temperature, jacket temperature, and smoke density. In addition to logging these items, the proximate and ultimate analyses of the fuel and the combustible in the ash and refuse were determined. These data are sufficient to permit the computation of the principal losses in a conventional heat balance, namely, heat lost in steam in flue gases, heat lost in dry flue gases, heat lost in carbon monoxide, and heat lost in combustible in ash. These losses constitute the stack and grate losses. The output is expressed in Btu per hour and was obtained by subtracting the total stack and grate losses from the heat value of the coal used.

This method may be utilized in the case of anthracite and coke where unaccounted losses due to unburned hydrocarbons in the flue gases are not present.

It is fair to assume an allowance for these losses in those tests where high volatile fuels are used. A value of 20 per cent was used. Data presented in several Bureau of Mines publications[2, 3] indicate this to be of the correct order.

Prior to the capacity tests, a preliminary run was made to determine the region of maximum heat exchanger temperature, and a thermocouple affixed at a representative point. Thermocouples were attached to the jacket, or outer casing, on each side panel at a height midway between the grate level and the top of the heat exchanger.

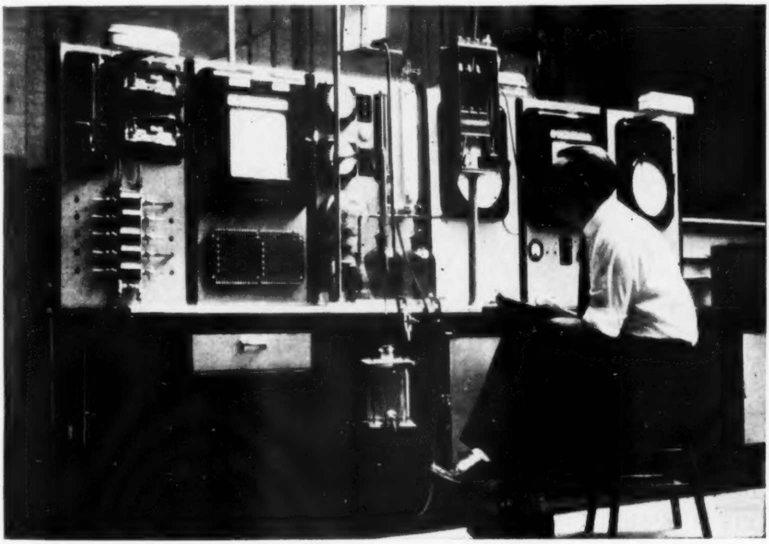

FIG. 2. TEST INSTRUMENT SET-UP

A preliminary starting fire was used in all tests. During the preliminary period and the test period the ashpit damper was adjusted to provide a maximum rate of burning with the following restrictions: (1) Maximum permissible flue gas temperature 900 F. (2) Maximum permissible heat exchanger temperature 1000 F. (3) Maximum permissible jacket surface temperature 400 F.

A draft of 0.02 in. water at the flue gas outlet was maintained constant in all runs. This is a representative value for the type of flue usually encountered in the field. Because the rate of burning was regulated by adjustment of the ashpit damper, however, the output is not strictly the result of the maintenance

[2] Heat Transference and Combustion Tests in a Small Domestic Boiler, by John Blizard, W. M. Myler, Jr., J. K. Seabright and C. P. Yagloglou. (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923.)
[3] Heat Transference and Combustion Tests in a Small Domestic Boiler, by H. W. Brooks, M. L. Orr, W. M. Myler, Jr., and C. A. Herbert. (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925.)

TABLE 1—TEST DATA

GENERAL INFORMATION					
1. Number of Test	1	2	3	4	5
2. Date of Test	7–9–42	7–16–42	7–22–42	7–30–42	8–4–42
3. Length of Test-Hour	11.0	12.5	6.0	4.0	8.0
HEATER DATA					
4. Heating Surface—sq ft	....	....	9.03	....	....
5. Grate Area—sq ft	....	....	0.80	....	....
6. Ratio Heating Surface to Grate Area	....	....	11.26:1	....	....
FUEL					
7. Kind	Anthracite	Coke	Bituminous	Lignite	Bituminous
8. Seam or Type	#6, D.L.&W.	Solvay	Illinois No. 6	Velva, N. D.	Island Creek
9. Size	Nut	Nut	2 in. x 3 in.	2 in. x 4 in.	2 in. x 3 in.
PROXIMATE ANALYSIS, AS FIRED, PER CENT					
10. Moisture	1.5	3.5	8.2	17.9	1.8
11. Volatile Matter	4.1	1.2	33.9	35.3	38.1
12. Fixed Carbon	83.6	88.3	50.5	41.2	55.5
13. Ash	10.8	7.0	7.4	5.6	4.6
14. Calorific Value, Btu per lb	12,880.0	12,850.0	12,235.0	8,952.0	14,190.0
ULTIMATE ANALYSIS, AS FIRED, PER CENT					
15. Carbon	76.6	87.7	68.5	37.6	79.6
16. Hydrogen	4.1	0.8	3.6	7.1	5.6
17. Oxygen	6.0	2.2	17.8	48.6	7.7
18. Nitrogen	1.5	1.3	1.6	0.6	1.3
19. Sulphur	1.0	1.0	1.1	0.5	1.2
20. Ash	10.8	7.0	7.4	5.6	4.6
WEIGHTS—POUNDS					
21. Total Weight Loss	27.5	16.25	24.25	29.25	24.0
22. Equivalent Fuel Burned	30.85	17.45	26.20	30.85	25.2
ASH AND REFUSE					
23. Weight of Ash and Refuse—lb	5.6	2.5	1.25	0.50	2.0
24. Combustible in Ash and Refuse—per cent	40.5	51.2	50.0	0.0	42.5
25. Carbon Burned per lb Fuel Fired—lb	0.69	0.80	0.66	0.38	0.76

DRAFTS—INCHES WATER										
26. Ashpit	0.010		0.020		0.018		0.020		0.018	
27. Flue Gas Outlet	0.019		0.021		0.021		0.020		0.020	
FLUE GAS COMPOSITION, PER CENT										
28. Carbon Dioxide (CO_2)	13.4		9.5		11.2		11.5		11.1	
29. Oxygen (O_2)	6.1		11.5		7.7		5.5		7.5	
30. Carbon Monoxide (CO)	0.5		0.0		0.5		3.9		0.1	
31. Nitrogen (N_2) (by Difference)	80.0		79.0		80.6		81.1		81.3	
32. Dry Flue Gas per lb of Fuel, lb	12.6		21.0		14.2		6.2		17.5	
33. Excess Air, per cent	38.3		123.0		56.6		20.2		53.1	
TEMPERATURES—F										
34. Air for Combustion	84.0		89.0		89.0		87.0		87.0	
35. Flue Gases	572.0		345.0		645.0		605.0		628.0	
HOURLY RATES										
36. Fuel Burned—lb per hour	2.8		1.39		4.37		7.71		3.15	
37. Fuel Burned—lb per sq ft Grate per hour	3.5		1.74		5.46		9.63		3.94	
38. Heat Release Rate—Btu per hour	36,064.0		17,861.00		53,367.00		69,019.00		44,698.00	
HEAT BALANCE	Btu	Per cent	Btu	Per cent	Btu	Per cent	Btu	Per cent	Btu	Per cent
39. Heat Lost in Steam in Flue Gases	471	3.7	83	0.6	424	3.5	816	9.1	648	4.6
40. Heat Lost in Dry Flue Gases	1,480	11.5	1,290	10.0	1,894	15.5	772	8.6	2,270	16.0
41. Heat Lost in Carbon Monoxide	252	2.0	0	0.0	286	2.3	977	10.9	69	0.5
42. Heat Lost in Combustible in Ash	1,022	7.9	1,022	8.0	292	2.4	0	0.0	526	3.7
43. Total Measurable Losses	3,225	25.1	2,395	18.6	2,896	23.7	2,565	28.6	3,513	24.8
44. Unaccounted Losses (Assumed)	0	0.0	0	0.0	2,447	20.0	1,790	20.0	2,836	20.0
45. Calorific Value of Fuel	12,880	100.0	12,850	100.0	12,235	100.0	8,952	100.0	14,190	100.0
46. Heat Utilized, Efficiency	9,655	74.9	10,455	81.4	6,892	56.3	4,597	51.4	7,841	55.2
OUTPUT										
47. Output—Btu per hour	27,034		14,539		30,116		35,476		24,673	
48. Heat Transfer Rate—Btu per sq ft per hour	2,995		1,610		3,335		3,930		2,735	
SMOKE										
49. Average concentration	0		0.5		—		—		—	
50. Average Ringelmann Number	0		1−		1+		1+		1+	
51. Average min. per hour No. 1 Ringelmann, or Greater	0		2		15		16		32	
52. Average min. per hour No. 2 Ringelmann, or Greater	0		1		14		15		16	
53. Average min. per hour No. 3 Ringelmann, or Greater	0		0		12		14		13	
54. Average min. per hour No. 4 Ringelmann, or Greater	0		0		11		14		11	
55. Average min. per hour No. 5 Ringelmann	0		0		9		6		10	

of a constant draft of 0.02 in. water. The test period was considered at an end when the flue gas temperature had dropped to 75 per cent of the value maintained during the preliminary period.

TEST RESULTS

Complete test data are shown in Table 1, and the heat balances are shown graphically in Fig. 3. The test with anthracite shows an output of 27,034 Btu per hour at 74.9 per cent over-all efficiency, the principal heat balance loss being in the dry flue gases. Fig. 4 is a graphical log of operation. It shows a stable temperature condition and reasonably constant CO_2. A slight amount of smoke, less than No. 1 Ringelmann, was found at the start of the test.

The lower reactivity of high temperature coke had a definite effect upon test results obtained with this fuel. Although a high efficiency, 81.4 per cent, was

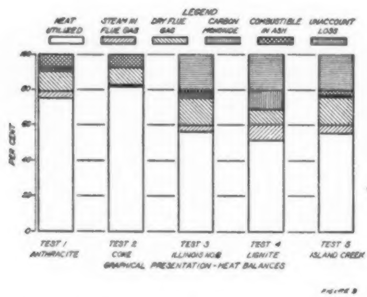

FIG. 3. GRAPHICAL DIAGRAM OF HEAT BALANCES

attained, the fuel burning rate and output were considerably lower than with anthracite. A short interval of smoke was recorded at the start of the test. The graphical log of this test, Fig. 5, shows a very stable flue gas temperature and CO_2.

The test with the free-burning Illinois No. 6 coal gave an output of 30,116 Btu per hour at 56.3 per cent efficiency. The burning rate was exceeded only by lignite. The principal measureable loss was the dry flue gas loss, and it will be noted that this is less than the assumed 20 per cent unaccounted loss. Fig. 6 shows the erratic flue gas temperature caused by changes in fuel bed condition, ignition of the over-fire gases, and manual attention. A similar trend is found in the CO_2 record.

Very bad smoke was produced in this test, No. 5 Ringelmann density persisting for over 30 min at the start. The smoke greatly exceeded permissible values established by municipal smoke ordinances.

The principal characteristics of the lignite test are the high rate of combustion, and the high losses due to carbon monoxide in the flue gases, and the steam in the flue gases. The lignite used in this test had been in storage for

several months and analysis showed 17.9 per cent moisture as compared to 37 per cent, as mined.

The over-all efficiency was only 51.4 per cent, but because of a much higher rate of burning than with the other fuels, the output was greater. The test

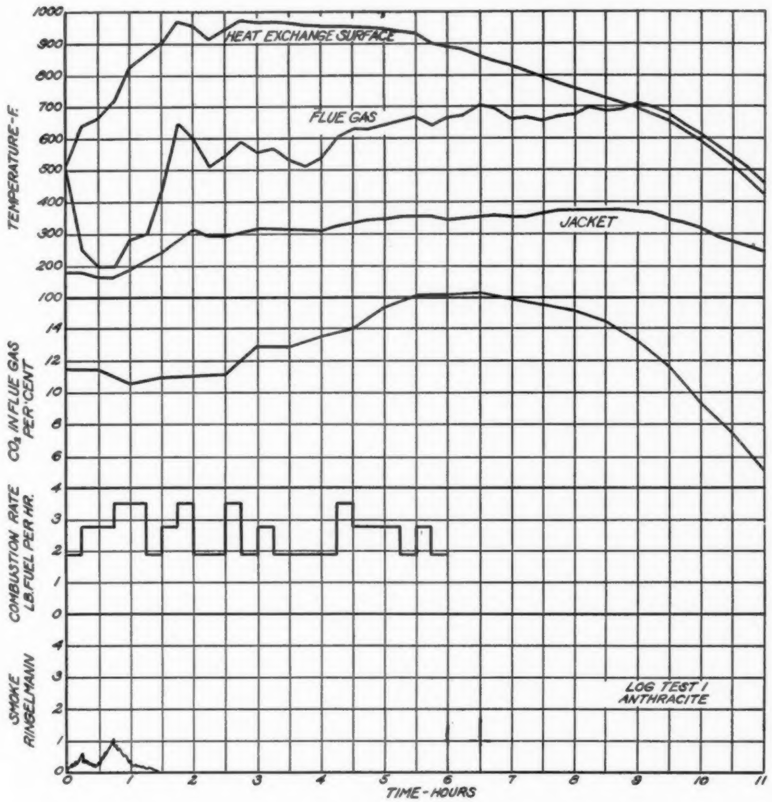

FIG. 4. GRAPHICAL LOG OF ANTHRACITE TEST

log with lignite, shown in Fig. 7, shows a period of approximately 45 min of No. 5 Ringelmann smoke at the start of the test.

The test run with a coking bituminous coal, Island Creek, gave an output of 24,673 Btu per hour at 55.2 per cent efficiency. The similarity in the heat balances for this fuel and the Illinois No. 6 coal is noted. The lower output for the Island Creek coal is principally due to the lower burning rate. Fig. 8

shows the graphical log for this fuel. The smoke produced in this test was the greatest of all the fuels.

A general review of the data shows that good combustion was obtained with all fuels except the lignite, where high *CO* was found.

Referring to the temperature limitation values established, it is shown that the permissible rate of output was governed largely by the heat-exchanger

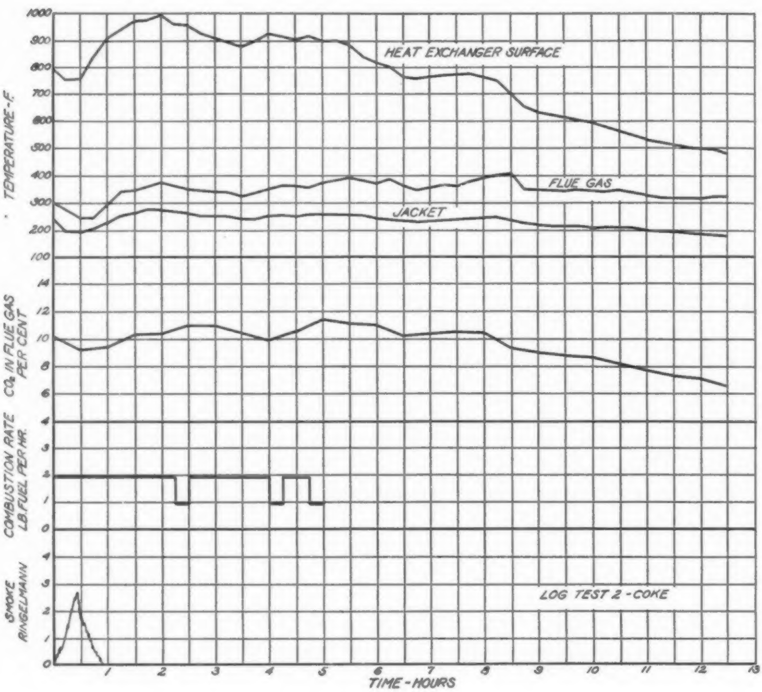

FIG. 5. GRAPHICAL LOG OF COKE TEST

limitation of 1000 F, and the maximum jacket temperature of 400 F. A limiting flue-gas temperature of 900 F was attained only in the test with Island Creek coal.

This series of tests offers evidence confirming the belief that capacities obtained with anthracite are fairly applicable to those for bituminous coals. Excepting coke and lignite, following output relationships were found:

Anthracite	100 (unity)
Bituminous (free burning)	111
Bituminous (coking)	91
Average variation from unity	10.0 per cent

The formation of dense smoke for long periods of time when firing high volatile fuels emphasizes the need for the development of improved types of heaters that will be free from this defect. At the present time much attention is being given to this problem by Battelle Memorial Institute and others.

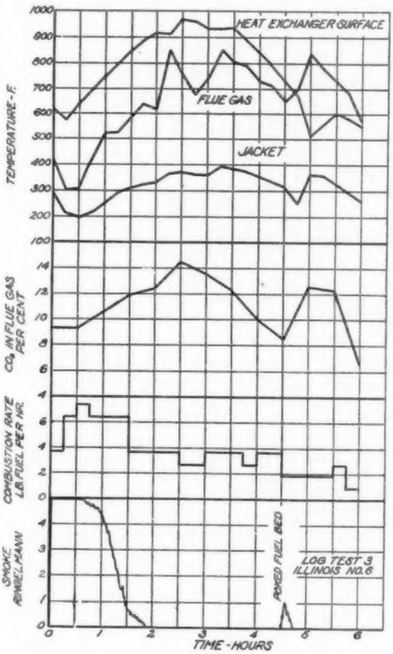

FIG. 6. GRAPHICAL LOG OF ILLINOIS NO. 6

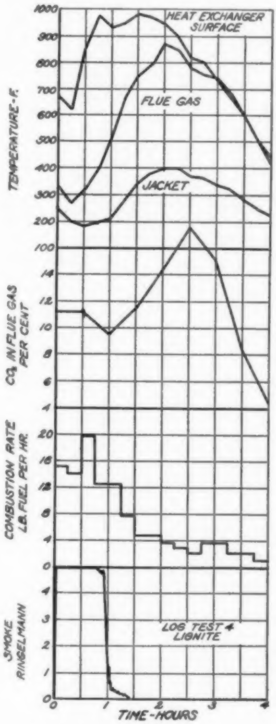

FIG. 7. GRAPHICAL LOG OF LIGNITE TEST

TENTATIVE RATING METHOD

Inasmuch as the heater used was fairly representative of its type, and the data are comprehensive, the following method of rating was evolved from the anthracite test. It is understood that this is a suggested method *only*, and confirming evidence from additional tests by other laboratories would be most helpful in evaluating the method.

The method is based upon a formula similar to that promulgated by The National Warm Air Heating and Air Conditioning Association [4] in which the

[4] The Standard Gravity Code, *National Warm Air Heating and Air Conditioning Association.*

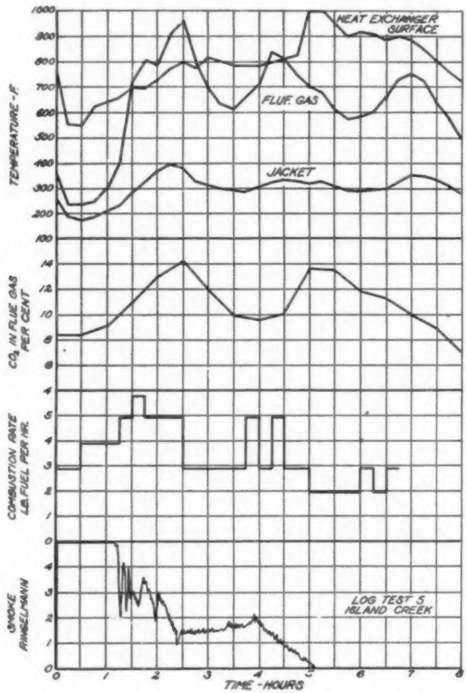

FIG. 8. GRAPHICAL LOG OF ISLAND CREEK COAL

several physical dimensions and relationships of a warm air furnace are employed.

A study of the ratio of heating surface to grate area of a number of typical gravity circulating heaters for solid-fuel firing showed a mean ratio of 16:1. This value is used in the basic formula:

$$\text{Output} = 4 \times 12{,}500 \times 0.75 \times \frac{G}{144}[1 + 0.02(R - 16)]$$

$$= 260 \times G[1 + 0.02(R - 16)]$$

Where

Output = Btu per hour
G = Grate area, square inches
R = Ratio heating surface to grate area

This equation is based on a combustion rate of 4 lb coal per square foot grate per hour, calorific value of fuel 12,500 Btu per pound as fired, and an over-all efficiency of 75 per cent. A 2 per cent allowance is made for each unit above or below the basic 16:1 ratio of heating surface to grate area.

Applying the proposed rating formula to the heater used in the test program, the following data are obtained:

$$\text{Output (Btu per hour)} = 260 \times 115.2[1 + 0.02(11.26 - 16)]$$
$$= 27{,}256$$

This is compared to the output obtained by test with anthracite, which was 27,034 Btu per hour.

It is understood that the output obtained by the above method is a *gross output.* In order to provide a proper pickup and overload factor, the maximum Btu per hour loss of the heated space should not exceed 80 per cent of the gross output.

CONCLUSIONS

The material in this discussion has been presented as an interesting example of the comparative performance of a specific heating device when fired with a wide variety of solid fuels. It is hoped that it has served a useful purpose in throwing some light upon a field in which little technical information is generally available.

DISCUSSION

H. R. LIMBACHER, Columbus, Ohio: Many and varied types of heating stoves have been perfected since the days of Franklin, but few have been carefully tested. The public is becoming improved-heater-conscious. It is necessary to know the performance of an ordinary stove before one can know how much better a so-called improved-heater operates.

The test procedure for the tests presented was much like that proposed in the Tentative Commercial Standard for Coal-Burning Space Heaters, proposed by the *National Bureau of Standard,* with the exception of the limit on stack draft which they set at 0.060 in. water. If that draft ceiling had been used instead of 0.020 in. water the fire would have responded more quickly after firing and the maximum permissible temperatures could have been more nearly maintained with a resultant increase in heat output. The higher stack draft would have caused more air leakage into the heater over the fuel bed and would have aided in completing combustion, thereby reducing the loss due to CO in the flue gas.

The author does not state the method used in firing the bituminous coals. An 18-in. firepot is large enough to permit some manipulation of the fuel bed at the time of firing, and the smoke emission might have been decreased by a change in firing method.

The log of the tests shows well the trend of the smoke emitted but the smoke data presented in Table 1 has little significance. Many smoke ordinances allow smoke for a period of 6 min in any one hour, but no consideration is given for an average duration per hour, however small.

R. L. DAVISON, New York, N. Y.: What height of chimney do you figure for that draft?

MR. CROSS: I will answer that question in this way: it is a very pitiful situation. The organization with which I am connected sells heaters of this type, and other types of heating equipment, to a very widespread class of people, and we run into many different situations. I have seen this type of heater repeatedly in low bungalows where there would be about 10 ft of smoke pipe running around the room and there

was a height of not more than 8 ft from the take-out in the wall to the top of the chimney.

To go on to Mr. Limbacher's comment as related to this particular phase, it is recognized that the tests reported in this paper do not conform to the draft standards set forth in Bureau of Standards TS 3297. I defend the premise of using 0.02 in. draft for two reasons: we had previously run some tests at 0.02 in. draft which we wished to check. These tests constitute a part of that check; furthermore I believe that 0.06 in. draft is an optimum figure. There are many installations that are operated with a lower available draft than that.

In regard to the effect of the higher draft, the introduction of additional air would have doubtlessly increased the CO_2, accelerated the rate of burning, and increased the output, but that would have been somewhat offset by the increased flue gas temperature which would tend to lower the efficiency, and the output.

It is true that an 18-in. firepot does permit some manipulation of the fuel bed, such as utilizing the side-bank method of firing, but the average user of this type of heater will not follow such practice, even if properly instructed. Straight over-feed firing methods were used in all tests in order to conform with expected customer usage.

The smoke data are not presented to indicate relationships with the various smoke ordinances, but it is believed that they do indicate the relative smoke-producing tendencies of the several fuels.

No. 1229

OPERATION OF THE RESEARCH HOME WITH REDUCED ROOM TEMPERATURES AT NIGHT

By A. P. Kratz,* W. S. Harris,† and M. K. Fahnestock,‡ Urbana, Ill.

In the broadest sense the object of the investigation in the Research Home[1] is to study, under actual service conditions, the different types of steam and hot-water heating systems, and various changes therein, from the standpoints of the systems and their component parts and the environment produced within the building. The specific object of the tests discussed in this paper was to determine the fuel and power savings and room temperature conditions resulting from operating with reduced indoor temperatures at night.

Description of Research Home and Heating System

The Research Home and heating system have been described in a previous paper.[2] The Research Home is a two-story building typical of the small, well-built American home. The construction is brick veneer on frame, and all of the outside walls and the second story ceiling are insulated with mineral wool bats 3⅝ in. thick. All windows and outside doors are weather stripped, and during the past two winters two storm doors were used. The calculated coefficient of heat transmission, U, for the wall section is 0.074 Btu per square foot per hour per degree Fahrenheit, and the calculated heat loss under design conditions for the house, excluding the basement, is 43,370 Btu per hour with temperatures of -10 F outdoors and 70 F indoors. The total floor area of the heated space is 1,174 sq ft and the volume is 9,393 cu ft. The house was completely equipped with thermocouples, recording thermometers, humidity indicators and recorders, and all other instruments necessary in making observations on tests.

The heating plant consisted of a one-pipe, forced-circulation, hot-water system, installed in connection with a three-section, cast-iron, oil-burning boiler, having a net I=B=R rating of 63,000 Btu per hour. In open recesses below the windows 261 sq ft of small-tube type, 19-in., 4-tube, cast-iron radiators were used. Based on a heat transmission rate of 200 Btu per square foot per

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[1] The I-B-R Research Home in Urbana, Ill., was built, furnished, and completely equipped for research work in steam and hot-water heating by the *Institute of Boiler and Radiator Manufacturers* in 1940.

[2] Performance of a Hot-Water Heating System in the Research Home, by A. P. Kratz, M. K. Fahnestock, W. S. Harris, and R. J. Martin. (A.S.H.V.E. Transactions, Vol. 48, 1942.)

Presented at the 49th Annual Meeting of the American Society of Heating & Ventilating Engineers, Cincinnati, Ohio, January, 1943.

hour, with a mean water temperature of 195 F, and an allowance for recessing, this was just sufficient to offset the calculated heat loss from the house.

METHOD OF TESTING

During all tests reported in this paper the operation of the heating plant was controlled by a heat anticipating thermostat located 30 in. above the floor in the living room. An oil burning rate of 1.1 gal per hour was used. This rate was the minimum obtainable with a clean fire and with a CO_2 content of the flue gases of not less than 8 per cent at the smoke outlet of the boiler. The oil used weighed 7 lb per gallon and had a calorific value of 19,550 Btu per pound.

The doors between rooms were open at all times. At 7:00 a.m., 11:00 a.m., 3:00 p.m. and 7:00 p.m. observations were recorded of the room air temperatures as determined by the thermocouples located 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling. Complete daily records were made of the operating time, the number of cycles and the power consumption of the oil burner and circulator, and of the weight of oil consumed. Recording instruments made continuous records of the stack temperature and draft, the CO_2, the temperature of the water at the boiler outlet and return, the outdoor air temperature, the indoor relative humidity, and the air temperature in each room 3 in. and 30 in. above the floor and 3 in. below the ceiling. Other daily observations included the total amount of electricity, gas and water used in the house, the number of occupants, and general weather conditions.

The heating system was arranged so that it could be operated either with or without a flow control valve and low limit aquastat. With the flow control valve in the system the water was circulated at a rate of about 13 gal per minute while the circulator was in operation. When the circulator was not in operation, the flow control valve prevented any circulation of water taking place between the boiler and the radiators. Without the flow control valve in the system, the water was circulated at the same rate while the circulator was in operation; but when the circulator was not in operation some circulation of water could occur by gravity action. In all cases the operation of the circulator was controlled by the room thermostat.

In order to determine whether the same amount of fuel saving could be effected by reducing the indoor temperature at night with each of these two methods of operation, a total of four series of tests was run. Two of these series, *A* and *B*, were run during the heating season of 1940-41 and the results have been discussed in a previous paper.[3] Two additional series, *C* and *D*, were run during the heating season of 1941-42. The test conditions for the different series as related to the setting of the thermostat and the status of the flow control valve and low limit aquastat are given in Table 1.

RESULTS OF TESTS

Fuel Consumption and Operating Cycles: Curves comparing the burner performance for series *A* in which the house was maintained at a uniform temperature of 72 F 24 hr per day, and series *C*, in which the setting of the thermostat was reduced to 66 F at 10:00 p.m. and the thermostat was reset to

[3] Loc. Cit. Note 2.

72 F at 5:30 a.m., are shown in Fig. 1. With a given average outdoor temperature over a period of 24 hr, any saving in fuel consumption resulting from reducing the thermostat setting at night is brought about by the fact that the average indoor temperature, and therefore the average indoor-outdoor temperature difference is reduced as compared with that obtained when the room air temperature is maintained at 72 F for 24 hr. With a given heating system and house, the average indoor-outdoor temperature difference for the 24 hr determines the

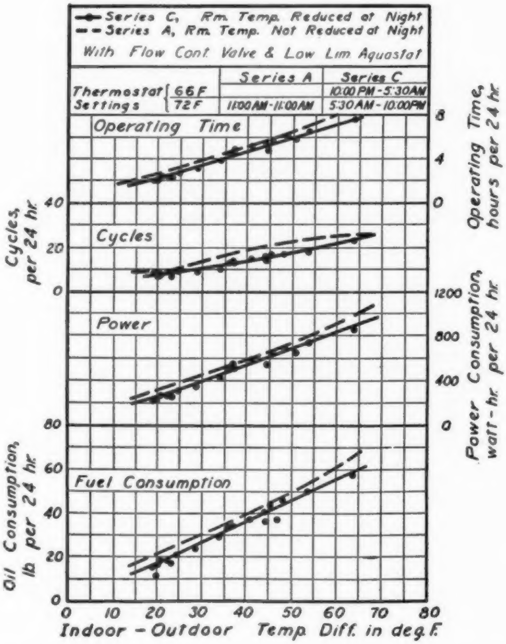

Fig. 1. Burner Performance Curves, Series A and C

heat loss from the house irrespective as to whether this difference was obtained with a constant indoor temperature for the 24 hr as in series *A*, or with a variable one, as in series *C*. Hence these two series cannot be compared on the basis of a 24-hr average for the indoor-outdoor temperature difference as applied to both series, because at a given difference the same fuel consumption would probably be obtained, and the result would be one curve instead of two. In order to determine the fuel saving, it was necessary to make comparisons for days that had the same average outdoor temperature over the 24 hr, with the further condition that the same indoor temperature was maintained during the daytime for both series of tests. These conditions were satisfied by basing the indoor-

outdoor temperature differences for series *C* and *D* on the average indoor temperature between 11:00 a.m. and 10:00 p.m. and the average outdoor temperature for 24 hr, and for series *A* and *B* on the 24-hr averages for both the indoor and outdoor temperatures. All of the curves in Figs. 1 to 4 inclusive have therefore been plotted on this basis.

For both series *A* and *C* the heating system was equipped with a low limit aquastat and flow control valve. Operation with the thermostat setting lowered to 66 F at night resulted in reductions in the burner operating time, cycles of

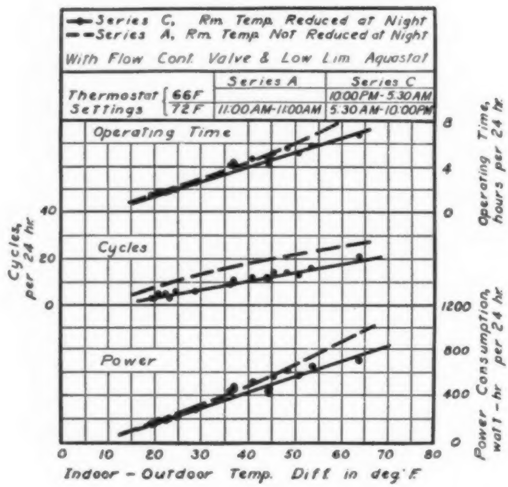

FIG. 2. CIRCULATOR PERFORMANCE CURVES, SERIES A AND C

burner operation, burner power, and fuel consumption throughout the complete range of outdoor temperatures occurring during the tests. Savings effected at an indoor-outdoor temperature difference of 34 F are approximately representative of seasonal savings. At an indoor-outdoor temperature difference of 34 F the actual saving in fuel amounted to 10.4 per cent.

TABLE 1—TEST CONDITIONS

THERMOSTAT	WITH FLOW CONTROL VALVE[a] AND LOW LIMIT AQUASTAT	WITHOUT FLOW CONTROL VALVE AND LOW LIMIT AQUASTAT
Set at 72F during 24 hours..	Series *A*	Series *B*
Set to 66 F at 10:00 P.M. and to 72 F at 5:30 A.M.......	Series *C*	Series *D*

[a] Low limit aquastat set for minimum temperature of 165 F.

At any given indoor-outdoor temperature difference occurring within the range of weather conditions encountered, fewer cycles of burner operation were obtained in series *C* than in series *A*. The greatest difference in the number of cycles accompanied temperature differences of from 40 to 50 F. With temperature differences of less than 40 F the rate of cooling of the air in the house was such that the thermostat would operate the burner not more than once or twice during the night. When the indoor-outdoor temperature

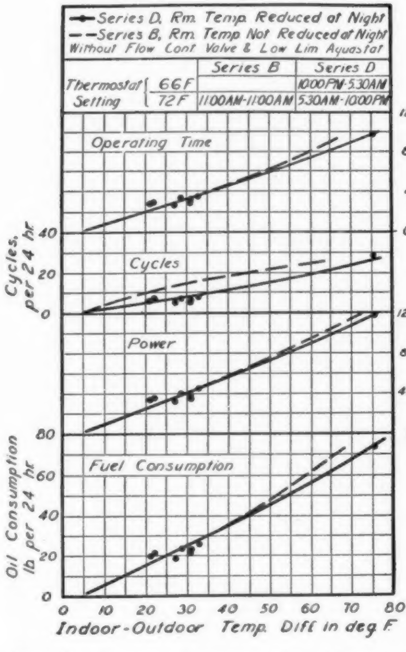

FIG. 3. BURNER PERFORMANCE CURVES, SERIES B AND D

difference was greater than 50 F the rate of cooling increased and sufficient time was required to raise the air temperature to 72 F in the morning, so that the high limit aquastat caused the burner to cycle several times. As a result, the number of burner cycles in series *C* approached that obtained in series *A* as the indoor-outdoor temperature difference became greater than 50 F.

Fig. 2 shows the circulator performance curves for series *A* and *C*, plotted on the same basis as the burner performance curves in Fig. 1. At a 34 F indoor-outdoor temperature difference the operating time of the circulator was 5.5 per cent less for series *C* than it was for series *A*. The curves in Figs. 1 and 2 indicate that, as the indoor-outdoor temperature difference increased, an

increase in the percentage of fuel saving and circulator and burner operating time occurred.

Operating under the conditions of series *C* resulted in a slightly greater reduction in the number of cycles of operation for the circulator than for the oil burner. By comparing the curves representing the burner and circulator cycles for series *C*, in Figs. 1 and 2, it may be observed that at all indoor-outdoor temperature differences two or three more cycles of operation were obtained for the burner than for the circulator. During the night, in order to keep the temperature of the water in the boiler above that corresponding to the

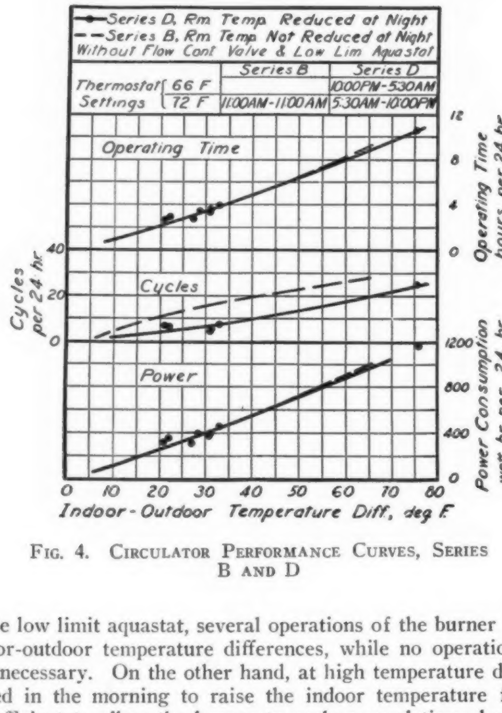

FIG. 4. CIRCULATOR PERFORMANCE CURVES, SERIES B AND D

setting of the low limit aquastat, several operations of the burner were required at low indoor-outdoor temperature differences, while no operation on the circulator was necessary. On the other hand, at high temperature differences, the time required in the morning to raise the indoor temperature from 66 F to 72 F was sufficient to allow the burner to cycle several times by action of the high limit aquastat. During this time, since the circulator was controlled only by the room thermostat, it operated continuously.

Comparisons of the burner and circulator performances obtained for series *B* with those obtained for series *D* are shown in Figs. 3 and 4. During both of these series the heating system was operated without the flow control valve and low limit aquastat, and for series *D* the setting of the room thermostat was reduced at night.

The curves show that at, and below, an indoor-outdoor temperature difference of 34 F, reducing the indoor temperature at night resulted in no saving

in the fuel used, or in the operating time and power consumption for the burner and circulator. The fuel saving indicated at an indoor-outdoor temperature difference of 34 F is only approximately representative of seasonal savings and since the curves do indicate some savings at indoor-outdoor temperature differences greater than 45 F, it is reasonable to assume that some seasonal saving would be effected. However, since only a relatively small portion of the total heating season is represented by indoor-outdoor temperature differences greater than 45 F, it is also evident that in the case of operation without the flow control valve and low limit aquastat in the system no material seasonal saving resulted when the indoor temperature was reduced at night. On the other hand, over the whole range of indoor-outdoor temperature differences, a reduction

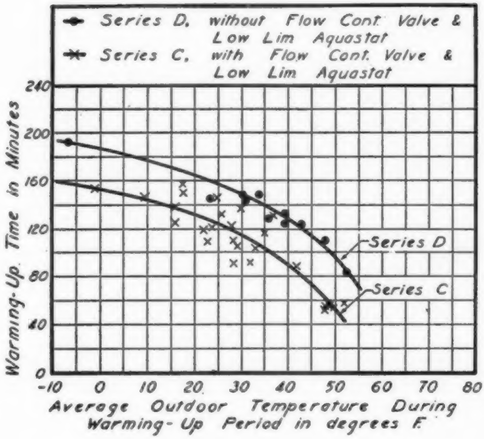

FIG. 5. TIME REQUIRED FOR MORNING WARMING-UP PERIOD

in the number of operating cycles for both the burner and circulator was obtained. This reduction was approximately the same as that exhibited by the results for series *C* as compared with those for series *A*.

Over the whole range of outdoor temperatures, the time required to raise the average house temperature from 66 F to 72 F as shown in Fig. 5, was approximately 30 min less in series *C* than it was in series *D*. In the discussion in the section on Water Temperatures it is shown that this reduction of 30 min in the time for warming up in the morning is a direct result of the additional heat stored in the water in the boiler when operating the system with a flow control valve and low limit aquastat. The additional 30 min of operation during the warming-up period in series *D* was almost equal to the difference in total saving in burner operating time resulting from the reduction of indoor temperature at night in series *C* as compared with series *D*, and was sufficient to account for the fact that the saving effected in series *D* was less than that

effected in series *C*. At an average indoor-outdoor temperature difference of 34 F, the burner operating time and fuel consumption were approximately the same in series *B*, *C* and *D*.

Water Temperatures: When the thermostat setting was reduced at night there was always a long off-period following the time at which the change was made. With the flow control valve in the system, series *C*, the water in the radiators and uninsulated piping cooled to room temperature during this long off-period, while that in the boiler cooled at an average rate of about 10 F per hour until the temperature was approximately 165 F at the location of the low limit aquastat. When this occurred, the burner operated for a sufficient

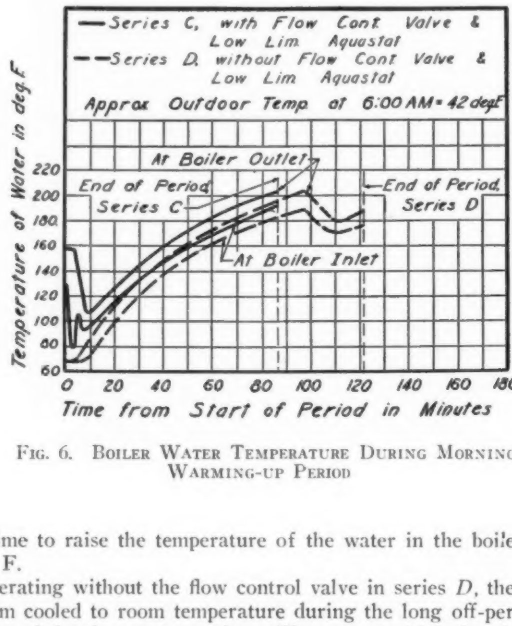

FIG. 6. BOILER WATER TEMPERATURE DURING MORNING WARMING-UP PERIOD

length of time to raise the temperature of the water in the boiler to approximately 175 F.

When operating without the flow control valve in series *D*, the water in the entire system cooled to room temperature during the long off-period following the reduction of the thermostat setting. The rate at which the air in the house cooled at night was such that, for the average winter day, the thermostat would not start the burner during the entire period of from 10:00 p.m. to 5:30 a.m. Under these conditions, in both series *C* and *D*, at the start of the morning warming-up cycle the temperature of the water in the radiators and piping was about the same as the indoor air temperature. In series *D* the water in the boiler had also cooled to room temperature, while in series *C* it could never cool below approximately 165 F.

For both methods of operation, with and without flow control value and low limit aquastat, the curves in Fig. 6 show the variations in temperature of the water at the outlet and inlet connections of the boiler during the entire warming-up period occurring after the thermostat had been reset to 72 F. The

outdoor temperature was approximately 42 F at 6:00 a.m. in both cases. The effect of the presence of the hot water in the boiler at the start of the warming-up period in series *C* is shown by the full line curves. The temperature of the water at the boiler outlet was 158 F during the first 4 min of the period, after which it dropped rapidly to about 108 F and then gradually increased until it attained a value of 203 F at the end of 85 min. At this time the indoor air temperature reached 72 F, and the room thermostat stopped both the burner and the circulator. For series *D*, when the outdoor temperature was 42 F, the temperature of the water at the boiler outlet, represented by the upper broken line, was only 68 F at the start of the warming-up period. This temperature steadily increased until it reached 204 F at 97 min after the start of the period. At this time the high limit aquastat stopped the burner, but not the circulator. The outlet water then cooled to 180 F, at which time the burner was restarted and continued to operate for the remainder of the warming-up period. In series *D* the time required to raise the indoor air temperature to 72 F was 121 min, or 36 min longer than that required with operation on series *C* at the same outdoor temperature.

Over the total time of the warming-up period the heat released in the house by the radiators and piping is directly proportional to the area between the inlet and outlet boiler water temperature curves. Since the rate of circulation of the water through the heating system during the on-periods of the circulator was the same in both series, the proportionality factor would also be the same, and the heat releases for the two series can be directly compared by comparing areas included between the respective inlet and outlet temperature curves. Over the total time included in the warming-up period for each series, there is less than 2 per cent difference between the total areas obtained from the inlet and outlet water temperature curves representing series *C* and those representing series *D*. This indicates that, at a given outdoor temperature, the same quantity of heat was supplied to the house during the warming-up period by both methods of operation. However, in series *D* not only the water in the radiator and piping, but also the boiler castings, refractories and the water in the boiler cooled to room temperature at night. As a result, during the additional 36 min required to bring the air temperature to 72 F, the burner operated 22 min in order to supply the extra heat necessary to bring the boiler castings, refractories and the water in the boiler back to normal temperature in the morning. Since, at a given outdoor temperature, the same amount of heat was supplied to the house during the warming-up period for both series *C* and *D*, but 22 min more of burner operation were required for series *D* than for series *C*, it seems evident that a lower overall efficiency was obtained in series *D* than in series *C*. Part of the heat represented by the 22 min of additional burner operation was used to bring the boiler castings, refractories, and the water in the boiler back to normal temperature, and the remainder appeared as increased chimney losses resulting from the higher flue gas temperatures accompanying the longer period of burner operation occurring in series *D*.

Room Temperature Gradients: The indoor air temperature differentials obtained with series *C*, operating with reduced indoor air temperature at night, and series *A*, operating with a constant indoor air temperature of 72 F maintained for 24 hours, are shown in Fig. 7. The temperature differentials were based on the avetages of the temperatures in all of the rooms. For the purpose of determining whether the reduction in indoor temperature at night and the

subsequent morning pick-up in series *C* had any effect on the indoor air temperature differentials obtained with normal day-time operation, only the temperatures prevailing during the period over which the house was maintained at 72 F were considered as having any significance. Hence, for series *C* in Fig. 7, the room temperature differentials obtained from the average indoor temperatures from 11:00 a.m. to 10:00 p.m. for a given day were plotted against the indoor-outdoor temperature difference obtained by subtracting the average temperature at the 30-in. level between the hours of 11:00 a.m. and 10:00 p.m. from the average outdoor temperature for the 24 hours. From Fig. 7 it may be observed that the reduction in the indoor temperature at night

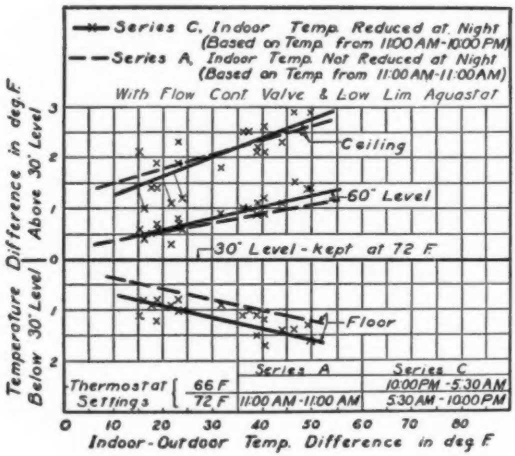

FIG. 7. INDOOR AIR TEMPERATURE DIFFERENTIALS

in series *C* had no significant effect on the indoor air temperature differentials.

At all indoor-outdoor temperature differences the temperature at the floor was about 0.3 F lower in series *C* than in series *A*. Between the running of series *A* and *C*, two sections were added to one of the radiators in the living room and five sections were removed from the radiator in the SW bedroom. After these changes were made in the plant, a few check tests, run under the conditions of series *A*, indicated that the slight change in the temperature of the air at the floor should be attributed to causes other than any effect resulting from the reduction in room air temperature at night. The curves shown in Fig. 7 also proved to be characteristic of the operation without the flow control valve and low limit aquastat in series *B* and *D*.

A typical graphic log showing air temperature variations and times of burner and circulator operation in the Research Home when operating with reduced indoor temperatures at night is presented in Fig. 8. At 10:00 p.m. the thermostat was set at 66 F. From 10:00 p.m. to 12:30 a.m. neither the burner nor

the circulator operated, while the air temperature at the 30-in. level decreased from 72 F to 66 F. From 12:30 a.m. to 5:30 a.m. the burner and circulator operated intermittently to maintain the air temperature at 66 F. At the latter time the thermostat was set at 72 F, and the burner and circulator started. The circulator operated continuously for 2.5 hours, during which time the house temperature increased from 66 F to 72 F. In the course of this 2.5 hour warming-up period the oil burner operated continuously for the first 75 min, and the temperature of the water in the system increased until the high limit aquastat stopped the burner. In about 10 min the water cooled enough to permit the aquastat to again start the burner, after which, under control of the

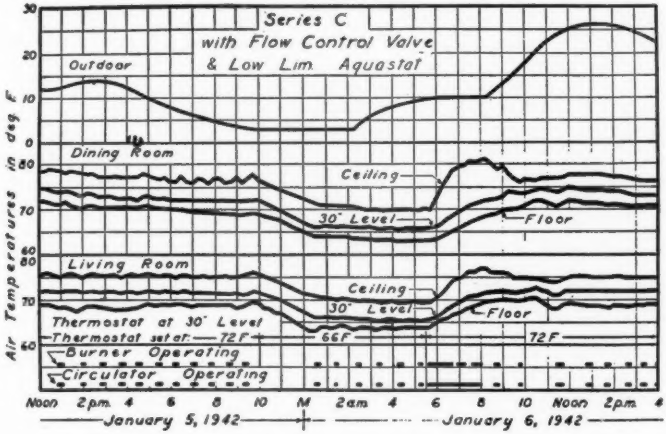

FIG. 8. GRAPHIC LOG FOR OPERATION WITH REDUCED INDOOR TEMPERATURES AT NIGHT

high limit aquastat, the burner operated intermittently until the end of the 2.5 hour warming-up period.

The air temperatures for both the dining room and the living room have been included in Fig. 8 since the overruns in temperature obtained in these two rooms during the warming-up period were the maximum and minimum observed in any of the rooms of the house. At the end of the warming-up period at 8:00 a.m., the temperature at the 30-in. level did not exceed 72 F in either the dining room or living room. On the other hand, the temperature 3 in. below the ceiling in the dining room reached a maximum of 81 F while that in the living room attained a maximum of only 77 F. At the same time the temperature of the air 3 in. above the floor in the dining room was 68 F and that in the living room was 69 F. Thus it is evident that, while some overrun was exhibited in the temperature of the air 3 in. below the ceiling when the burner and circulator were stopped at the end of the warming-up period, no overrun in temperature occurred at the control level, 30 in. above the floor.

After the warming-up period, the natural circulation of air tended to equalize the temperatures at the different levels, decreasing the temperature of the air

at the ceiling and slightly increasing that at the floor and control levels, so that by about 10:00 a.m. the normal temperature differentials had been established in the rooms. Similar records to those shown in Fig. 8 were made of the temperature variations in each room of the house. These records indicated that there was a rough correlation between the amount of overrun in the temperature of the air near the ceiling and the ratio of the room volume to the amount of radiation installed. As this ratio increased, the amount of overrun decreased.

After changing the setting of the thermostat from 72 F to 66 F at night, the rate at which the temperature of the air in the rooms dropped was primarily dependent upon the outdoor temperature prevailing during the cooling period.

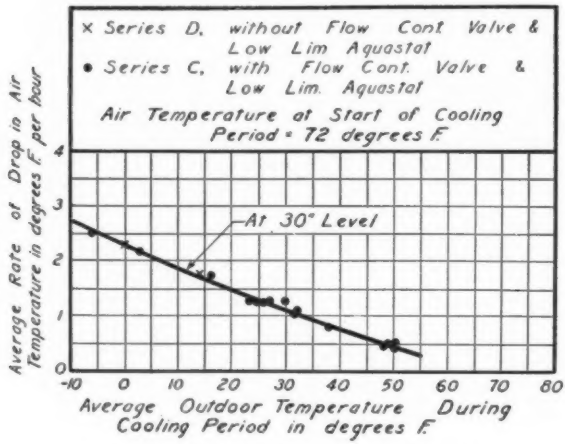

FIG. 9. RATE OF COOLING OF LIVING ROOM

The average rate of temperature drop for the living room, which is also representative of that for the house as a whole, is shown in Fig. 9. An indoor temperature of 66 F was never attained during the 7.5 hours between 10:00 p.m. and 5:30 a.m. unless the average cooling rate was equal to or greater than $\frac{6}{7.5}$, or 0.8 F per hour. From Fig. 9 it may be observed that this occurred only when the average outdoor temperature at night was 39.5 F or less. It is therefore evident that at outdoor temperatures above 39.5 F, setting the thermostat back more than 6 F would not result in any additional saving in fuel over that which could be obtained by the 6 F setback.

The cooling rate is a function of house construction and the indoor-outdoor temperature difference, and the total fuel saving that can be effected by operating with reduced indoor air temperatures at night is largely a function of three factors: (1) The number of degrees the thermostat is set back at night.

(2) The number of hours of operation on reduced temperature at night, and (3) the rate at which the house cools. Taking into consideration the length of time required to warm the house in the morning and the time required to overcome the effect of cold walls and furniture, it is probable that, except in the most severe climates, the practical limits for night setback of the thermostat lie somewhere between 6 and 10 F.[4]

Summary

The following is a summary of the results obtained in this investigation:

1. By setting the thermostat to 66 F at 10:00 p.m. and restoring the setting to 72 F at 5:30 a.m. a saving of approximately 10 per cent was effected in the seasonal fuel consumption and burner operating time, and approximately 5.5 per cent saving in circulator operating time, when operating the system with a flow control valve and low limit aquastat maintaining a minimum boiler water temperature of 165 F.

2. Only a slight saving in seasonal fuel consumption, burner operating time and circulator operating time was effected when the thermostat was set back at night and the system was operated without a flow control valve and low limit aquastat. This condition probably resulted from the fact that it required a longer time to warm the house in the morning with this method of operation than that required when operating with a flow control valve and low limit aquastat, and the additional losses during this time tended to offset the possible saving effected by the night set-back.

3. Exclusive of the warming-up period in the morning, reducing the indoor temperature at night had no effect on the temperature conditions within the house during the day.

4. While overruns of from 2 to 5 F were observed in the temperature of the air at the ceiling during the morning warming-up periods, no such overruns occurred at the control level 30 in. above the floor.

5. In average winter weather the indoor air cooled not more than 6 F during the 7.5 hours that the thermostat was set back at night.

Acknowledgments

The results presented in this paper were obtained in connection with the investigation of steam and hot water heating systems in the Research Home at the University of Illinois, conducted by the Engineering Experiment Station, of which M. L. Enger, Dean of the College of Engineering, is director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the Institute of Boiler and Radiator Manufacturers and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is hereby made to R. J. Martin, formerly Special Research Assistant, for services rendered in collecting and reducing test data. Acknowledgment is also made to the various manufacturers who cooperated by furnishing materials and equipment.

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[4] Investigation of Oil-Fired Forced-Air Furnace Systems in the Research Residence, University of Illinois Engineering Experiment Station *Bulletin* 318, pp. 49-57.

No. 1230

COMPARATIVE RESISTANCE TO VAPOR TRANSMISSION OF VARIOUS BUILDING MATERIALS

By L. V. Teesdale,* Madison, Wis.

Introduction

For several years the Forest Products Laboratory[1] has been working on a study of the problems of condensation of moisture within walls and roofs of homes during cold weather. Such condensation arises as a result of the migration of vapor from the interior of a building through intervening materials to some surface cold enough to be below its dew-point temperature where it may be converted into either water or frost. The presence of moisture as a result of the condensation within walls or attic spaces may lead to undesirable aftereffects which not only increase maintenance costs but may also shorten the life of the structure. The most practical method known at this time of protecting a structure against condensation is to place, on the warm side of the wall, materials having high resistance to vapor transmission.

Since the amount of vapor that can pass through a material is inversely proportional to the resistance of the material to vapor transmission, the first phase of the study was to determine the comparative resistance of various building materials to vapor transmission. Plaster, plywood, fiberboards, coated papers, and other membranes were accordingly used as vapor barriers in test panels installed in an experimental house. Later a special project was set up to make a comprehensive study on various paper products, and the results were published in the TAPPI Technical Association Papers, Series 22-1939, under the title, Comparative Resistance to Vapor Transmission of Commercial Building Papers. The transmission values in that report were given in grams lost per 24 hours per square meter, a method of measurement commonly used in the pulp and paper industry. The transmission values of the various papers were recalculated on a basis of grains lost per square foot per hour, a method of measurement more familiar to architects, engineers, and others connected with the construction industry. In addition to the values for the various paper products, the work here reported includes transmission values for plywood, fiberboards, and numerous miscellaneous building materials.

Experimental Procedure

The method of test used is similar to the vapometer test as used by the pulp and paper industry but modified to adapt it to the materials being tested. The sample to be tested is sealed over a pan containing sufficient water to saturate the air space below the sample, and the pan is placed in a room maintained at

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[1] Maintained at Madison, Wisconsin, in cooperation with the University of Wisconsin.

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a constant temperature of 80 F and humidity of 30 per cent. A weight-time record is kept from which a curve is plotted showing the weight of vapor lost through the sample at each weighing. The curve at equilibrium becomes a straight line, the slope of which gives the equilibrium transmission rate.

Apparatus and Method

The pan is made of sheet copper 10 x 10 in. square and 2 in. deep with a ¾-in. shoulder and ¾-in. rim. Calking compound or roofer's cement, gun consistency, is spread about $\frac{1}{16}$ in. thick on the shoulder and a thin covering spread over the inner and outer edge of the rim. About 1 in. of water is placed in the pan. A layer of about ¼ in. of sphagnum moss is also added to reduce the tendency of the water to surge during handling of the pan and also to add evaporative surface to the water. The test sample, previously cut or shaped to a size of 11¼ x 11¼ in. is slipped into place and forced against the compound. A thin layer of compound is placed around the edge of the sample to act as an adhesive for the final covering of strips of aluminum foil. This foil, 0.001 in. thick, is mounted on wax paper also 0.001 in. thick and extends from the bottom of the rim outside of the pan up over the rim and down, covering the compound, and out over the sample about ¾ in., leaving an exposed surface on the sample of exactly 100 sq in.

The pan is placed in a room maintained at 80 F and 30 per cent relative humidity. After weighing, the pan is placed in a rack of shelves so spaced that there is about 3 in. of clear space over the sample surface. Oscillating fans agitate the air in the room, but the air velocity over the surface is only about 25 fpm. This condition was chosen in order to obtain the greatest uniformity and comparable conditions on all pans. Weighings were made daily in some cases, but with high resistance materials once each week, and continued for 5 weeks or more. Some samples were held under test for 3 months.

Often the transmission rate varied during the early part of the period of observation, but, with very few exceptions, the rate of loss eventually became constant. Materials like plywood, for example, showed a gradually increasing rate of loss for the first few days until an equilibrium moisture content condition was established through the thickness of the sample, after which the rate became constant. Some of the asphalt-coated papers showed a faster rate of loss for the first 2 or 3 days than thereafter. Possibly some of the oils used in the compound crept out into certain absorbent types of samples and accounted for the faster loss immediately after exposure that was shown in a few instances. Only those weights obtained after reaching a constant rate of loss were used in determining the resistance value for the sample.

Accuracy Obtained by Tests

Weighing: The error in weighing the test pans was less than 0.5 gram although the pans weighed between 2500 and 4000 grams. An average of five or more weights was used so that the error probably was less than 0.2 gram.

Sampling: The possible error resulting from samples not being representative of the material is unknown. Nevertheless duplicate runs on similar products generally indicated comparatively close results.

Efficiency of Seal: Check tests using sheets of tin in place of a permeable

substance showed a very small initial loss from the solvent in the compound after which the loss of weight in 9 weeks' time was negligible.

Temperature: The temperature in the room was maintained at 80 F. The variation of this temperature was never greater than ± 1 deg. The temperature within the pan, of course, followed that of the room.

Relative Humidity: The relative humidity in the test room was maintained at 30 per cent, ±2.5 per cent. The air under the sheet was kept essentially saturated at all times due to the presence of the water. Some deviation from complete saturation may have occurred where the samples had a fast rate of transmission.

Circulation: The velocity of air across the surface of the pans averaged 25 fpm. Sufficient circulation was maintained in the room to keep the humidity practically constant.

Suitability of Test

Vapor barriers, as used in building construction, are exposed to vapor pressure differences of less magnitude than those used in these tests. Moreover, vapor barriers in use are seldom exposed to 100 per cent humidity on one side. However, for all practical purposes it was assumed that the more severe exposure used in the test method would merely magnify any differences in materials and aid in differentiation. Comparison with results in the walls of the test house corroborate these ideas.

Some types of material, particularly the saturated and coated sheathing papers, sometimes showed a more rapid initial rate of transmission than the equilibrium rate. Other materials, such as plywood, had an opposite initial rate. Only the equilibrium rate of transmission was used in establishing a value for a sample.

Definitions and Terminology

Certain terms used in the pulp and paper industry have been used as a partial basis for classifying the papers tested and in discussing results. Since there is some variation in their usage, the terms are defined as they are used here:

Building paper is a general term for papers used in building, especially sheathing papers. The word *paper* is used to include *felts.*

Felts refer to a group of bulky, rough and cheap building papers; felts are ordinarily distinguished from sheathing papers in that they are more loosely formed and thicker.

Saturated is a term describing a sheet that has been more or less completely penetrated by a waterproofing liquid, usually by means of a dipping process, the excess liquid being removed leaving a dull surface on the finished sheet.

Infused is the manufacturer's term for a process of dipping whereby the sheet surface absorbs the waterproofing liquid with very little penetration.

Sheathing paper is a paper commonly used between the sheathing and siding boards in house construction. This class has several divisions: (1) *Tarred sheathing,* also called *slater's felt,* is a paper made usually of old stock (rags, wool, mixed papers, etc.) saturated with tar; (2) *asphalt sheathing* is a sheet from the same type of stock, but saturated with asphalt instead of tar; (3) *saturated and coated sheathing paper* is a sheet made from the same type of stock saturated and coated with asphalt. The coating is an asphalt of high melting point and frequently has a small amount of wax added.

Tarred felt is a felt saturated with tar.

Asphalt felt is a felt saturated with asphalt.

Thread felt or *string felt* is a felt with reinforcing strings.

Duplex paper is a paper composed of two sheets. A duplex building paper is usually composed of two kraft sheets joined by an asphalt layer, or *lamination*. A *reinforced* duplex paper contains strings, usually of jute or hemp, in the lamination to impart tear resistance. A *creped duplex paper* is one in which the sheets have been creped before lamination or one which has been creped after assembly.

In duplex papers it is common to designate the product by the weight of paper and lamination per ream ($24 \times 36 - 480$). For example, a 30/40/50 paper would be one containing 30 lb of paper as one sheet, 40 lb of asphalt lamination, and 50 lb of paper as the other sheet, on a standard ream basis.

Reflective surfaced or *silvered* refers to a deposit of metal or metallic oxides in some form on the surface of the paper to impart heat reflectance.

PRESENTATION AND DISCUSSION OF RESULTS

Table 1 is composed of several divisions, each containing the data on a certain class of material. Where paper products are sold in units of a different size, as, for example, roofing in rolls of 108 sq ft, the weight per 500 sq ft was calculated and entered in the table. The approximate retail price in 1938 for certain paper products was obtained either from local dealers or the manufacturers and quoted in terms of rolls of 500 sq ft. The tabulated values relating to the composition of materials are based upon questionnaires sent to the manufacturers. No inquiry was made as to their accuracy.

In judging the merit of a product, a tentative standard vapor transmission rate of 0.600 grain per square foot per hour has been employed. From data obtained in tests in the laboratory test house, as well as observations and experience, it appears that vapor barriers having such resistance and located under the lath or on the warm side of the wall offer sufficient protection in houses maintaining 40 per cent humidity in normal winter weather having short periods of sub-zero weather.

Single sheets of felt saturated with tar, commonly called tar paper, tarred felt, or slater's felt rank low in resistance to vapor transmission. Similar sheets saturated with asphalt are far more resistive though they fall far below the tentative resistance desired as a vapor barrier. These two classes of paper are better suited for use as sheathing papers where low resistance to vapor transmission is desired and where resistance to transmission of liquid water and to wind infiltration is also required.

Single sheets saturated and coated with asphalt have high resistance to vapor transmission as a class. It was noted that, in general, sheets having a high gloss were more resistant than those having a dull surface. Material in this class makes excellent material for vapor barriers. They are inclined to be brittle in cold weather and require some care in handling at that time.

Heavier single-sheet saturated felts classed as roll roofing have very high resistance to vapor transmission. Asphalt shingles are made of similar material and no doubt would also be equally high in resistance. Roll roofing generally is more expensive than the saturated and coated papers and would not be used in residence construction as a vapor barrier, but no doubt it could be used in other places where more severe service is expected, for example, in surfacing the inner lining of cow barns.

There are many manufacturers of duplex papers, and the products of only a few companies were tested. Since duplex products vary in respect to cover sheets and to weight of asphalt laminae, these factors were studied through a wide range. In general it appears that the weight of asphalt used is the deter-

TABLE 1—PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR

Company designation	Sample	Material	Remarks	Approximate price per roll 500 sq ft (1938)	Weight per roll 500 sq ft	Number of tests	Variation of test from average	Loss in grains per square foot per hour
	Number			Dollars			Percent	
		Single Sheets Sheathing and Roofing Products						
		Saturated with tar (tar felt)						
A	10	Slater's felt or tarred sheathing	53.2 percent saturated	1.25	32.0	3	0.6	9.020
B	40	25 pounds tarred sheathing	48.0 ......do	.94	25.0	3	1.4	12.650
B	41	String felt	50.0 ......do	2.24	44.0	3	.5	12.830
B	42	Slater's felt	40.0 ......do	1.08	30.0	3	2.5	9.815
B	55	25 pounds tarred felt	63.3 ......do	1.82	124.0	1	....	11.970
		Saturated with asphalt (asphalt felt)						
A	17	Insulating paper	42.0 percent saturated	1.90	25.0	3	8.3	2.380
A	20	Insulating paper	50.0 ......do	2.00	30.0	3	14.8	2.542
A	19	Insulating paper	47.0 ......do	2.25	35.0	3	8.9	2.802
A	21	Insulating paper	48.8 ......do	2.40	40.0	3	3.2	2.470
A	9	Insulating paper	44.5 ......do	2.80	45.0	3	6.1	3.970
A	14	Insulating paper	48.2 ......do	3.00	55.0	3	1.5	4.430
C	38	Waterproof building paper	43.2 ......do	1.16	25.0	3	12.6	12.840
C	39	Insulating paper	49.0 ......do	2.75	45.0	3	3.6	12.650
C	84	15 pounds asphalt felt	62.5 ......do	2.89	75.0	3	5.8	3.205
C	85	30 pounds asphalt felt	63.4 ......do	5.78	150.0	3	12.8	2.485
B	43	25 pounds asphalt sheathing paper	48.0 ......do	.94	25.0	3	4.2	6.204
B	44	30 pounds asphalt sheathing paper	40.0 ......do	1.08	30.0	3	1.9	6.100
		Saturated and coated with asphalt (sheathing paper)						
D	1	Coated sheathing paper	....	2.90	35.0	3	6.8	.727
A	15	..........do..........	....	2.25	35.0	3	16.0	.308
A	16	..........do..........	....	3.00	50.0	3	16.0	.158
C	37	..........do..........	....	2.80	35.0	3	17.6	.456
C	36	..........do..........	....	3.40	50.0	3	3.0	.592
B	45	..........do..........	....	2.24	35.0	3	6.3	.197
B	18	..........do..........	....	3.01	50.0	3	8.5	.364
E	56	..........do..........	....	2.20	35.0	3	18.7	.296
F	24	..........do..........	....	....	50.0	3	13.6	.407
G	119	..........do..........	....	2.01	35.0	3	27.0	.160
G	118	..........do..........	....	3.50	50.0	3	9.5	.259
H	70	..........do..........	....	....	38.0	3	0	.160
H	69	..........do..........	....	....	50.0	3	6.4	.191
B	64	..........do..........	....	3.01	50.0	3	1.8	.222
		Saturated and coated felts (roll roofing)						
I	130	Heavy saturated felts	....	4.40	125.0	2	2.1	.864
F	82	..........do..........	....	3.62	125.0	3	1.8	.099
E	81	..........do..........	....	3.70	157.0	3	0	.105
C	136	..........do..........	....	5.50	208.0	3	21.5	.185
C	137	..........do..........	....	11.50	254.0	3	25.6	.080
C	83	..........do..........	....	13.50	301.0	3	9.1	.068

Duplex or Laminated Products. Untreated Kraft Asphalt Laminae Not Reinforced

J	138	30-30-30	.88	15.6	3	2.6	.957
J	146	30-30-30	.88	15.6	3	10.0	1.012
J	147	30-40-30	.98	17.4	3	12.2	.561
J	139	30-50-30	1.03	19.1	3	2.1	.487
J	148	30-50-30	1.03	19.1	3	3.1	.456
J	149	30-60-30	1.12	20.8	3	3.0	.512
J	150	30-75-30	1.26	23.4	3	5.2	.357
J	140	30-95-30	1.45	26.9	3	1.4	.345
J	151	30-100-30	1.49	27.8	3	2.9	.364
K	164	30-30-60		20.9	2	13.8	.784
K	163	60-30-60		26.1	3	8.9	.345
K	165	30-60-30		20.9	3	4.0	.357
K	166	30-80-30		24.4	3	10.0	.290
K	167	30-100-30		27.8	3	10.4	.296
L	60	25-40-25	1.32	17.0	3	10.4	.592
L	59	30-45-30	1.50	18.0	3	36.9	.753
L	58	30-50-60	2.00	25.0	3	2.4	.777
L	57	60-50-60	2.62	30.0	3	13.0	.475
M	53	30-45-30	2.00	17.5	3	9.5	2.600
M	97	30-55-30	1.35	20.0	3	11.7	.846
N	30	30-30-30	1.62	15.5	3	27.0	3.530
O	25	30-70-30	2.50	22.4	3	8.7	.636
P	75	30-30-30		15.6	1		1.560

Duplex or Laminated Products, Untreated, Creped

N	29	Creped duplex	2.35	20.0	3	5.6	2.150
N	31	Creped duplex	3.50	42.0	3	4.0	.364
J	153	30-30-30 before creping	1.77	20.9	3	4.0	4.120

Duplex or Laminated Products, Untreated, Reinforced

O	6	30-80-30	5.00	29.0	3	7.9	1.018
Q	7	30-85-30	4.75	29.0	3	29.0	1.270
R	33	30-100-60	3.80	34.0	3	.7	.900
M	52	30-193-30		47.0	3	22.4	.660
L	61	30-110-30	3.75	33.0	3	10.9	1.690
L	62	30-80-30	2.62	25.0	3	12.4	.598
L	63	30-70-30	2.12	23.0	3	3.7	1.005
P	74	30-95-30		31.0	3	40.6	4.440
M	93	30-90-30	3.00	28.0	3	16.1	.574
M	95	30-70-30	3.00	25.0	3	6.7	.550
S	115	30-70-30	2.85	25.0	3	11.0	.784
R	35	Three sheets 30-pound kraft two layers asphalt 50 pounds	4.30	34.0	3	5.7	.722

Treated Duplex or Laminated Products. One or More Sheets Saturated with Asphalt

J	152	30-30-30 saturated both sides	.97	17.7	3	10.0	1.492
J	154	30-30-30 saturated both sides and creped	1.94	23.6	3	13.3	2.250
J	155	35-50-35 saturated both sides and creped	2.42	29.4	3	3.0	6.230
R	32	50-125-50 saturated both sides and creped	5.50	40.0	3	1.2	1.570
P	73	45-90-45 saturated two sides, reinforced		31.0	3	7.3	4.150
E	91	Heavy duplex -- one sheet untreated	1.72	90.0	3	4.1	.407
S	116	Three sheets -- two layers asphalt and reinforcing	4.50	33.0	3	8.4	.876
I	129	30-100-30 reinforced	3.50	32.0	3	11.3	.982

Duplex Sheets with Reflective Coating -- Asphalt Laminae

M	51	Coated wall paper	15.00	61.0	3	9.2	.401
M	54	Coated fabric, reinforced	17.50	55.0	3	19.1	.388
M	92	Coated double (30-60-30), reinforced	12.50	25.0	3	3.6	1.862
M	135	Coated double (30-60-30), reinforced		56.7	3	2.9	.357

TABLE 1—PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR—*Continued*

Company designation	Sample	Material	Remarks	Approximate price per roll 500 sq ft (1938)	Weight per roll 500 sq ft	Number of tests	Variation of test from average	Loss in grains per square foot per hour
	Number			Dollars			Percent	
		Duplex Sheets with Miscellaneous Treatments and Coatings						
M	8	30-85-30, reinforced, coal tar saturated in cover sheets		5.00	31.4	3	19.5	1.498
M	26	30-75-30, not reinforced	do	3.00	25.5	3	3.8	1.130
M	94	30-105-30, surface coated with asphalt and wax		4.50	32.0	3	2.6	.407
M	96	30-130-30, reinforced with burlap		5.50	53.0	3	21.9	2.240
T	104	30-25-30 waxed one side		1.08	15.0	3	4.1	.494
T	105	Do	stripped from insulation	....	....	3	7.5	.536
		Plain Kraft Building Papers						
U	28	Rosin sized sheathing paper	....	....	20.0	3	1.8	49.200
U	27	Do	....	....	40.0	3	11.0	39.600
J	142	Do	....	....	5.2	3	3.6	46.800
J	143	Do	....	....	10.4	3	.7	43.200
J	158	Do	Two sheets 30 pounds kraft	....	10.4	3	.5	40.500
J	159	Do	Two sheets 60 pounds kraft	....	20.8	3	.8	36.900
		Papers Used in Commercial Packaging						
V	3	Kraft soaked with wax (paraffin)		....	21.2	3	10.2	9.880
V	4	Do		....	22.5	3	1.3	21.800
V	5	Do		....	25.0	3	4.5	13.320
W	133	Insulating paper used for wrapping		2.24	19.1	3	4.0	.469
X	132	40 pounds kraft coated with synthetic elastic material		....	....	3	3.3	2.405
Y	124	20 pounds paper with attached layer of waxy material		....	....	2	3.6	.432
Y	125	40	do	....	....	2	6.0	.259
Y	126	64	do	....	....	1	....	.277
		Insulation back-up papers--surface treated only						
T	47	Single infused, new, ribbed		....	14.5	3	3.7	1.990
T	72	Do		....	14.5	1	....	2.283
T	77	Do		....	14.5	3	1.9	.648
T	108	Do		....	14.5	3	25.6	3.455
T	12	Single infused, stripped from insulation, ribbed		....	14.5	3	4.6	.935
T	78	Do		....	14.5	3	3.6	1.204
T	120	Do		....	14.5	3	32.8	.827
T	106	Double infused, new, ribbed		....	15.0	3	8.1	2.655
T	87	Double infused, stripped from insulation, ribbed		....	15.0	3	38.0	2.320
Z	......	Stripped from insulation		....	....	2	....	.649
Z	......	Stripped from insulation, ribbed		....	....	2	....	10.520
AA	......	Back-up paper, stripped from insulation, ribbed		....	....	3	....	1.850
BB	......	Do		....	....	3	....	.620
CC	......	Do		....	....	2	....	1.962
T	101	Double infused without ribs		....	10.0	3	1.8	1.675
T	102	Do		....	12.0	3	24.8	1.141
T	103	Do		....	14.0	3	11.3	2.621
T	98	Do		....	12.0	3	22.8	1.515
T	99	Do		....	16.0	3	42.1	3.143
T	100	Do		....	24.0	3	7.2	3.540

Single sheet--kraft--double infused

T	46	Building paper, light	....	....	12.0	3	3.2	1.940

Single sheet--kraft--double infused

T...	48	Building paper, light	....	12.0	3	3.2	1.940
T...	49	Building paper, medium	....	16.0	3	43.1	1.333
T...	50	Building paper, heavy	....	24.0	3	7.2	3.540
G...	117	Black kraft paper	....	14.3	3	4.4	6.960

<u>Miscellaneous Pulp and Paper Products</u>

DD...	......	Insulation, multiple corrugated board 1 inch thick, lightweight kraft	....	....	2	....	21.080
EE...	......	Ground newsprint bound with asphalt and composition 1 inch thick	....	....	3	....	27.400
FF...	......	2-1/8 inch corrugated boards coated, two-coated reflective surfaces	....	....	3	....	12.845
GG...	......	Corrugated board saturated with asphaltic oil	....	....	3	....	8.500
HH...	......	Asphalt board 0.694 pound per square foot 0.102 inch thick, soft felt face	....	....	3	....	.180
HH...	......	Asphalt board 0.817 pound per square foot 0.118 inch thick, hard texture face	....	....	3	....	.092
HH...	......	Asphalt board 0.838 pound per square foot 0.125 inch thick, soft felt face	....	....	3	....	.308

Company designation	Thickness	Species	Glue	Remarks	Number of tests	Loss in grains per square foot per hour: Low	High	Average
				Plywood				
				Commercial plywood vegetable glue				
II...	Three-ply 1/4-inch	Douglas-fir	Soybean	No coating	1	......	......	4.500
Do...	...do...	...do...	...do...	One coat asphalt paint	3	2.340	2.560	2.470
Do...	...do...	...do...	...do...	Two coats asphalt paint	3	.586	.678	.617
Do...	...do...	...do...	...do...	One coat aluminum paint	3	.985	3.960	1.660
Do...	...do...	...do...	...do...	Two coats aluminum paint	3	.555	1.420	.925
Do...	Five-ply 1/2-inch	...do...	...do...	No coating	2	1.920	1.970	1.940
Do...	...do...	...do...	...do...	One coat asphalt paint	3	.740	1.110	.986
Do...	...do...	...do...	...do...	Two coats asphalt paint	3	.246	.370	.290
Do...	...do...	...do...	...do...	One coat aluminum paint	3	.986	1.080	1.030
Do...	...do...	...do...	...do...	Two coats aluminum paint	3	.678	1.110	.845
Market...	Three-ply 1/4-inch	...do...	...do...	One coat shellac	3	5.670	5.840	5.770
Do...	...do...	...do...	...do...	One coat aluminum paint, two coats lead and oil paint	3	1.100	1.275	1.221
				Commercial plywood artificial resin glue				
II...	Three-ply 1/4-inch	Douglas-fir	Artificial resin	No coating	3	3.080	4.690	4.060
Do...	...do...	...do...	...do...	One coat asphalt paint	3	1.730	2.030	1.910
Do...	...do...	...do...	...do...	Two coats asphalt paint	3	.370	.617	.512
Do...	...do...	...do...	...do...	One coat aluminum paint	1	......	......	2.160
Do...	...do...	...do...	...do...	Two coats aluminum paint	3	.862	.986	.925
Do...	...do...	...do...	...do...	One coat prime, one coat flat wall paint	1	......	......	1.910
Do...	Five-ply 1/2-inch	...do...	...do...	No coating	3	1.970	2.410	2.190
Do...	...do...	...do...	...do...	One coat asphalt paint	3	.493	.986	.722
Do...	...do...	...do...	...do...	Two coats asphalt paint	3	.278	.432	.326
Do...	...do...	...do...	...do...	One coat aluminum paint	1	......	......	1.110
Do...	...do...	...do...	...do...	Two coats aluminum paint	2	.493	.710	.605
Market...	Three-ply 1/4-inch	...do...	...do...	One coat shellac	24	3.640	4.640	4.064
Do...	...do...	...do...	...do...	One coat aluminum, two coats lead and oil	9	.931	1.285	1.146
Do...	Three-ply 3/8-inch	...do...	...do...	One coat aluminum, two coats lead and oil	3	1.133	1.471	1.298
JJ...	Five-ply 1/2-inch	Poplar	...do...	No coating	2	.185	.400	.333
Do...	...do...	...do...	...do...	Two coats aluminum, two coats lead and oil 3 years exposure	1	......	......	.216

TABLE 1—PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR—*Continued*

			Various species of plywood and glue					
Laboratory	Three-ply 3/8-inch	Douglas-fir	Tego	No coating	3	1.170	2.840	1.790
Do	do	do	Bakelite	No coating	1			1.660
Do	do	do	Bakelite	One coat Duroseal, one coat wax	1			1.230
Do	do	do	Durez	No coating	1			.802
Do	do	do	Durez	One primer, two coats enamel	1			.617
Do	do	do	Urea Hotpress	No coating	1			3.390
Do	do	do	Blood albumin	No coating	1			1.660
Do	do	do	Casein	No coating	1			2.470
Do	do	do	Soybean	No coating	1			3.020
Do	do	Birch	Tego	No coating	6	2.160	4.060	3.080
Do	do	do	Bakelite	No coating	1			3.570
Do	do	do	Bakelite	One primer, two coats enamel	1			1.170
Do	do	do	Durez	No coating	1			1.920
Do	do	do	Durez	Two coats aluminum paint	1			.740
Do	do	do	Urea Hotpress	No coating	1			4.750
Do	do	do	Urea Hotpress	One coat primer, two coats flat wall paint	1			2.250
Do	do	do	Blood albumin	No coating	1			4.500
Do	do	do	Blood albumin	Two coats aluminum paint	1			1.050
Do	do	do	Tego	No coating	1			4.630
Do	do	do	Tego	One coat aluminum paint	1			4.140
Do	do	do	Casein	No coating	2	6.720	6.970	6.850
Do	do	do	Casein	Two coats aluminum paint	1			.863
Do	do	do	Soybean	No coating	1			6.850
Do	do	do	Soybean	One coat Duroseal and wax	1			3.350
Do	Five-ply 1/2-inch	do	Tego	No coating	1			1.480
Do	do	do	Casein	No coating	1			5.000
Do	Three-ply 3/8-inch	do	Tego	One layer or sheet of glue in each glue line	1			3.580
Do	do	do	Tego	Two layers or sheets of glue in each glue line	1			3.270
Do	do	do	Tego	Three layers or sheets of glue in each glue line	1			3.140
Do	do	do	Tego	Four layers or sheets of glue in each glue line	1			2.710
Do	do	do	Tego	Five layers or sheets of glue in each glue line	1			2.510
			Resin treated and bonded plywood					
Laboratory	Three-ply 3/8-inch	Douglas-fir	Tego	20 percent resinol 5995 in aqueous solution	2	.123	.185	.154
Do	do	do	Tego	40 do	2			.062
Do	do	do	None	40 do	2			.062
Do	do	Yellow birch	Tego	20 do	1			.555
Do	do	do	Tego	40 do	1			.123
Do	Three-ply 3/16-inch	Douglas-fir	Tego	Untreated	2	3.086	3.333	3.209
Do	Three-ply 3/12-inch	do	Tego	Untreated	2	3.654	2.776	2.715
Do	Three-ply 3/16-inch	do	Tego	Outer-ply treated 20 percent resinol center-ply untreated	2	.345	.395	.370
Do	Three-ply 3/12-inch	do	Tego	Outer-ply treated 20 percent resinol center-ply untreated	1			.172
Do	Three-ply 3/16-inch	do	Tego	Outer and center plies treated 20 percent resinol	2	.302	.337	.320
Do	Three-ply 3/12-inch	do	Tego	do	2	.290	.327	.308
Do	Three-ply 3/16-inch	Spruce	Tego	Untreated	1			4.030
Do	do	do	Tego	Outer-plies treated 35 percent resinoid XR 5995 in aqueous solution	2	.572	.648	[1] .610
Do	do	Aspen	Tego	Untreated	1			4.360
Do	do	do	Tego	Outer plies treated 35 percent resinoid, etc.	2	.391	.782	.586
Do	do	Cottonwood	Tego	Untreated	1			5.420
Do	do	do	Tego	Outer plies treated 35 percent resinoid, etc.	3	.150	.331	.276
Do	do	Poplar	Tego	Untreated	1			5.660
Do	do	do	Tego	Outer plies treated 35 percent resinoid, etc.	1			.663
Ii	Five-ply 1/2-inch	Douglas-fir	Resin bonded and	covered both sides with resin impreg paper	3	1.670	1.890	[2]1.760
Do	Three-ply 1/4-inch	do		do	3	2.525	2.840	[1]2.698
			Plain Wood Substance					
Laboratory	3/8-inch	Douglas-fir	Solid wood--no glue		1			9.310
Do	do	Ponderosa pine	Solid wood--no glue		1			11.150

[1] Treated faces only partially cured before assembling. The faces were compressed during assembly because the resin plasticized the wood and reduced its compressive strength below that of the untreated core. The thickness of the treated panels was reduced by about 25 percent when compared with the untreated samples.

[2] Did not reach stable rate of loss in 11 weeks; Rate based on last 3 weeks while for first 5 weeks the rate would be about 20 percent of that used here.

... rate of loss in 11 weeks) Rate based on last 3 weeks while for first 5 weeks the rate would be about 20 percent of that used here.

Company designation	Material	Remarks	Number of tests	Loss in grains per square foot per hour: Low	High	Average
	Fiberboard Products					
Market	1/2-inch fiberboard, used on walls, one prime coat		3	22.600	24.400	23.300
Do	1/2-inch fiberboard, used on walls, one prime coat, two coats flat wall paint		3	6.980	11.800	8.760
Do	do	one coat aluminum	3	.762	1.000	.861
Do	do, one coat flat wall		2	.676	.762	.705
Do	1/2-inch fiberboard, used on walls, one prime coat, two coats flat wall paint, two coats aluminum, two coats flat wall		1			.880
Do	1/2-inch fiberboard, used on walls, one prime coat, two coats flat wall paint, one coat semigloss		1			.688
Do	do	two coats semigloss	2	.616	.674	.645
Do	do	one coat aluminum, two coats semigloss	2	.806	.880	.843
Do	1/2-inch fiberboard, used on walls, one prime coat, two coats flat wall paint, two coats aluminum, two coats semigloss		2	.485	.600	.543
Do	1/2-inch fiberboard, used on walls, one prime coat, two coats flat wall paint, one coat proprietary sealing varnish, two coats flat wall		2	.732	.732	.732
Do	1/2-inch fiberboard, used on ceilings, one prime coat		3	26.620	27.430	27.200
Do	1/2-inch fiberboard, used on ceilings, one prime coat, two coats flat wall paint		3	2.605	3.315	3.100
Do	do	one coat aluminum, one coat flat wall	3	.880	1.470	1.125
Do	1/2-inch fiberboard, used on ceilings, one prime coat, two coats flat wall paint, one coat aluminum, two coats flat wall		3	.732	1.143	.894
Do	1/2-inch fiberboard, used on ceilings, one prime coat, two coats flat wall paint, two coats aluminum, two coats flat wall		3	.850	1.488	1.200
Do	1/2-inch fiberboard, used on ceilings, one prime coat, two coats flat wall paint, one coat semigloss		3	.718	1.113	.888
Do	do	two coats semigloss	3	.806	1.130	1.018
Do	do	one coat aluminum, one coat semigloss	3	.704	.936	.797
Do	1/2-inch fiberboard, used on ceilings, one prime coat, two coats flat wall paint, two coats aluminum, two coats semigloss		3	.558	.646	.612
Do	1/2-inch fiberboard, used on ceilings, one prime coat, two coats flat wall paint, one coat proprietary sealing varnish, two coats flat wall		3	.792	1.025	.889
KK	1/2-inch paper fiberboard, uncoated		3	13.000	13.360	13.150
Do	1/2-inch paper fiberboard, uncoated		3	17.200	20.200	18.370
CC	1/2-inch fiberboard insulating plaster base, uncoated		3	25.850	27.250	26.300
Do	1/2-inch fiberboard coated lath 30 pounds asphalt per thousand	Bottom[1]	2	9.121	9.744	9.432
Do	do 35 pounds do	Top[2]	2	3.200	3.350	3.272
Do	do 40 pounds do	Top[2]	2	2.318	2.390	2.354
Do	do 45 pounds do	Bottom	2	3.502	3.696	3.599
Do	do from test house	Top	2	2.750	3.730	3.240
Do	3/4-inch fiberboard sheathing asphalt coated both sides dull surface		2	2.190	3.641	2.960
LL	1/2-inch fiberboard insulating plaster base uncoated		4	13.150	27.380	21.960
Do	1/2-inch fiberboard asphalt coating, one side glossy surface		2	.435	.450	.442
Do	3/4-inch fiberboard sheathing uncoated		1			19.400
MM	1/2-inch fiberboard insulating lath, uncoated		3	24.600	25.200	25.000
Do	1/2-inch fiberboard wall tile, flat finish		3	25.600	26.400	25.870
Do	3/4-inch fiberboard sheathing asphalt surface, both sides aluminum coating		3	5.860	6.490	6.090
BB	3/4-inch fiberboard sheathing asphalt surface, both sides semigloss surface		2	.871	2.030	1.450

TABLE 1—PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR—*Continued*

Company designation	Material	Remarks	Number of tests	Loss in grains per square foot per hour: Low	High	Average
AA	1/2-inch insulating plaster base uncoated		3	25.820	26.900	26.450
Do.	1/2-inch fiberboard, wall tile flat finish		3	22.450	25.300	23.750
Do.	3/4-inch fiberboard sheathing asphalt coating, both sides dull surface		3	18.610	19.008	18.840
NN	1/2-inch insulating lath uncoated		1			23.800
Do.	1/2-inch insulating wallboard, plain		1			25.370
Do.	1/2-inch insulating wallboard, flat finish		1			27.420
Do.	3/4-inch insulating sheathing asphalt coating, both sides glossy surface		1			.743
OO	3/4-inch insulating sheathing asphalt coated covered with paper		1			1.132
Z	1/2-inch insulating lath uncoated		1			26.800
Do.	1/2-inch insulating wallboard uncoated, one side smooth		1			24.100
Do.	1/2-inch insulating wallboard coated, flat finish		3	15.630	17.400	16.320
Do.	3/4-inch insulating sheathing asphalt coating, both sides dull surface		1			4.880
	Plaster and Gypsum Products					
Laboratory	Plaster on wood lath		1			7.900
Do.	Plaster on wood lath, two coats aluminum paint		1			.331
Do.	Plaster on gypsum lath		1			13.550
Do.	Plaster on gypsum lath, one coat primer, two coats flat wall paint		1			2.680
Do.	Plaster on gypsum lath, one coat glue size, one varnish, one flat paint					2.620
Do.	Plaster on fiberboard lath					14.800
Do.	Plaster on fiberboard lath, one coat aluminum, two coats flat paint		1			2.680
Do.	Plaster on fiberboard lath, two coats lead and oil final glossy		1			3.000
Do.	Plaster Good wire lath backed up with foil covered paper					2.095
BB	Gypsum lath backed with aluminum foil		3	.132	.238	.189
Laboratory	Gypsum board, plaster base with paper removed both sides		2	32.900	35.100	34.000
Do.	Gypsum board, same painted with one coat interior primer and sealer		3	1.120	1.320	1.220
Do.	Plaster on gypsum lath painted with one coat interior primer and sealer		1			1.120
Do.	Gypsum board, painted one coat primer, one coat lead and oil		3	1.370	1.550	1.460
Do.	Plaster on bypsum lath, painted one coat primer, one coat lead and oil		1			1.870
Do.	Gypsum board, painted one coat primer, two coats lead and oil		3	.820	.924	.809
Do.	Plaster on gypsum lath, painted one coat primer, two coats lead and oil		1			1.352
Do.	Gypsum board, painted onecoat aluminum, two coats flat wall paint		3	1.246	1.353	1.301
Do.	Plaster on gypsum lath, painted one coat aluminum, two coats flat wall paint		1			.925
Do.	Gypsum board, painted one coat aluminum, two coats lead and oil		3	1.267	.903	1.139
Do.	Plaster on gypsum lath, painted one coat aluminum, two coats lead and oil		1			1.182
Do.	Gypsum board, painted two coats aluminum, two coats flat wall paint		3	.603	.752	.683
Do.	Plaster on gypsum lath, painted two coats aluminum, two coats flat wall paint		1			.430
Do.	Gypsum board, painted one coat primer, two coats lead and oil		3	1.223	1.246	1.239
Do.	Plaster on gypsum lath, painted one coat primer, two coats lead and oil		1			.925
Do.	Gypsum board, painted one coat proprietary sealing varnish, two coats flat wall paint		3	1.255	1.297	[3]1.276
Do.	Plaster on gypsum lath, painted one coat proprietary sealing varnish, two coats flat wall paint		1			[2]1.160
Do.	Gypsum board, painted one coat proprietary sealing varnish, two coats lead and oil		3	1.561	1.698	[1]1.632
Do.	Plaster on gypsum lath, painted one coat proprietary sealing varnish, two coats lead and oil		1			[1]1.335
Do.	Gypsum board, painted one coat glue size, two coats lead and oil		3	1.742	2.150	1.950
Do.	Plaster on gypsum lath, painted one coat glue size, two coats lead and oil		1			1.590
Do.	Gypsum board, painted one coat glue size, one coat aluminum, two coats flat wall paint		3	.946	1.142	1.070
Do.	Plaster on gypsum lath, painted one coat glue size, one coat aluminum, two coats flat wall paint		1			.856
Do.	Gypsum board, painted one coat glue size, one coat aluminum, two coats lead and oil		2	1.167	1.385	1.276
Do.	Plaster on gypsum lath, painted one coat glue size, one coat aluminum, two coats lead and oil		1			1.205
Do.	Gypsum board, painted one coat glue size, one coat primer, one coat flat paint		3	1.355	1.915	1.243
Do.	Plaster on gypsum lath, painted one coat glue size, one coat primer, one coat flat paint		1			1.074

[1]Applied to bottom as manufactured.
[2]Comparable with present material as supplied.
[3]Paint incompatability.

[1]Applied to bottom as manufactured.
[2]Comparable with present material as supplied.
[3]Paint incompatability.

	Miscellaneous Products				
Z	3/16-inch compressed fiberboard	1	....	....	3.640
Do	1/4-inch compressed fiberboard	1	....	....	5.550
PP	1/2-inch Vermiculite board bonded with asphalt	1	....	....	.269
CC	1/2-inch fiberboard faced one side with 1/8-inch sheet asbestos board	4	0.416	0.681	.556
Do	1-inch fiberboard faced two sides with 1/8-inch sheet asbestos board	3	.660	3.750	1.770
Do	1-inch cemented wood fiber and 3/8-inch ground newsprint bound with asphalt and composition	3	3.790	4.090	3.960
Do	Same with spray coat aluminum paint	3	3.060	3.720	3.377
Do	Same base plus troweled coat sand and cement asphalt emulsion and two coats aluminum paint	3	.396	.399	.398
OO	1-inch pressed cork board	1	....	....	4.440
QQ	3-5/8 inches rock wool	3	15.670	32.100	24.800
RR	Aluminum foil mounted one side kraft paper	3	.066	.136	.110
Do	Aluminum foil mounted two sides kraft paper	3	.066	.093	.085
M	Duplex sheet coated with oxides	3	....	....	1.080
Do	Duplex sheet coated with metal oxides, second test series	3	....	....	1.360
SS	Oiled and painted cloth used for wall coverings	3	39.200	39.800	39.400
Do	do	3	11.100	18.700	16.000
Do	do	3	3.240	3.420	3.300
Do	do	3	1.850	1.850	1.850
Do	do	3	2.590	3.700	3.330
Do	do	3	3.240	4.530	3.700
Do	do	3	2.770	4.530	3.820
TT	do	3	2.130	2.870	2.370
Do	do	3	2.040	3.790	3.080
Do	do	3	2.130	2.870	2.410
Do	do	3	2.410	2.410	2.410
Z	1/2-inch blanket double infused not ribbed, thick glossy asphalt layer on cover sheet	3	.594	.725	.649
Do	1-inch do	3	.725	.890	.780
Do	1/2-inch blanket single infused, ribbed not creped	3	2.225	9.860	7.288
Do	1-inch do	3	7.550	9.370	8.159
UU	1/2-inch and 1-inch blanket, early type 2-creped cone sheets	2	2.010	2.380	2.195
Do	1/2-inch and 1-inch blanket, late type do	9	1.160	1.728	1.400
Do	2-inch blanket early type do	1	....	....	1.422
Do	2-inch blanket late type do	3	1.098	1.225	1.168
Do	1/2-inch blanket, cover sheet not creped do	3	.652	1.314	.919
	Pan Tests				
Laboratory	Open pan without sphagnum moss	2	127.500	135.800	131.650
Do	Open pan with sphagnum moss	2	324.000	402.000	363.000
Do	Tin plate (in place of test sample)	2	0	0	0
Do	Caulking compound, covered as in test but no water	1	0	0	0
Do	Tin plate having one hole 0.0174 inch diameter	1	....	....	.0092
Do	Tin plate having one hole .0331 inch diameter	1	....	....	.0304
Do	Tin plate having one hole .0490 inch diameter	2	....	....	.0512
Do	Tin plate having one hole .0620 inch diameter	2	....	....	.0704

mining factor in obtaining resistance, particularly with plain papers. The 30/30/30 grades varied rather widely in resistance and several appeared to be somewhat below the desired resistance, but a 30/40/30 grade was generally acceptable and increasing the asphalt laminae above 50 lb per ream does not seem justified in terms of added resistance. Again a 30/60/30 grade seemed superior to a 30/30/60 grade, indicating the relative importance of the asphalt and the paper.

Creping the duplex papers seems to reduce the resistance greatly though in the case of the papers made by one manufacturer it had no effect.

Reinforcement added to duplex papers appears to lower the resistance to vapor transmission, and the cause might be attributed to the disturbance of the asphalt layer by the reinforcing material. It does not appear that reinforcement is necessary in duplex papers used as vapor barriers and the added cost and loss of resistance might well discourage its use in favor of the plain duplex materials.

The cover sheets of some duplex sheets are saturated, but the material used appears to act as a solvent for the asphalt laminae and lowers the final resistance of the sheet below that of unsaturated duplex papers of the same weight of paper and asphalt.

Reflective coatings applied to the surface of papers are of doubtful value in adding resistance, but often such coatings are applied over duplex papers and good resistances are obtained.

Plain rosin-sized kraft or building paper ranks very low in resistance, as might be expected. Some of the commercial packaging materials also rank rather low while others have very high resistance.

The insulation back-up papers appear to be variable in resistance and generally below the standard desired. Back-up papers used on blanket insulations appeared generally to be somewhat higher in resistance than those used on batts, but sometimes fell short of the required resistance. A house built in the summer of 1941 had blanket insulation of a type that is delivered in rolls. It was installed in the manner recommended by the manufacturer. During an extreme cold period early in January 1942, condensation appeared in the attic and side walls though some ventilation is provided in the attic. Other similar cases are reported from time to time.

While the Laboratory tests were under way the manufacturers were constantly working to increase the resistance values for the product now on the market.

SUMMARY OF TESTS OF PAPERS

The highest resistance in any class of papers tested occurs in the saturated and coated felts or roll roofing materials, followed by sheets saturated and coated with asphalt. In the latter type it seems that the resistance is obtained by the surface coating, and the total weight is not the important factor. The untreated duplex papers rank next in vapor resistance and appear to be more variable as a class than the saturated and coated products. The low price makes it attractive, and the fact that it does not become brittle in cold weather is also a desirable feature. It does have a tendency to shrink if alternately wet and dry and should be installed with some slack so that it will not tear or break with moisture changes. Those grades having 40 to 50 lb or more of asphalt in the middle lamina appear to be more consistent than where less asphalt is

used. It does not appear that reinforcement is necessary in vapor barriers; and since the reinforcement tends to lower the resistance to vapor transmission there is little excuse for their use as barriers. Saturating the cover sheets of duplex papers also appears undesirable as it seems to lower the resistance to vapor transmission without adding any desirable property for this class of service.

PLYWOOD

The vapor resistance of plywood is greater than that of solid wood of the same thickness and species. No doubt the glue line offers some resistance and possibly the breaking up of the continuity of grain from one face to the other is also a contributing factor. The resistance of plywood to vapor transmission increases with an increase in the thickness and number of plies. The kind of glue used appears to affect the resistance through a rather wide range, particularly in the artificial resin group. In only one case tested, however, was the resistance of the glue line high enough to equal the standard of resistance desired.

The species appears to affect the rate of vapor transmission, Douglas-fir showing definitely more resistance than birch in material matched for thickness and glue. Increasing the amount of glue in the glue line increases the resistance as shown where one, two, three, four, and five sheets of phenolic-resin glue were used in each glue line of 3/8 in. Douglas-fir plywood. Soaking the veneer before gluing in an aqueous solution of resinol increased the resistance to a remarkable degree.

The ordinary decorative coatings that may be applied to plywood are often of a type that are low in vapor resistance. Since coatings may be applied to the back side for vapor barriers, the type of finish coat on the face may be unimportant as a protection against vapor movement. Two coats of vapor-resistant paint seem to be required to obtain reasonable vapor transmission resistance. The first or priming coat often strikes into the plywood more or less, but the second coat generally finishes with a shiny surface. Asphalt paint seemed to offer somewhat greater resistance than aluminum paint.

FIBERBOARDS

Fiberboard products as used in building may be divided roughly into two classes, those intended for interior use either as a plaster base or as interior wall finish, and those used as a substitute for wood sheathing. They may be used for other purposes also, such as insulation, acoustical material, etc. As a plaster base the material usually is furnished 1/2 in. thick but may also be obtained 1 in. thick. As sheathing it is invariably 3/4 in., the same thickness as standard wood sheathing. As a material it has very low resistance to vapor transmission, though it may be obtained with coatings of asphalt that add more or less to the resistance depending upon the character of the coating. Fiberboards intended for interior wall finish may also be obtained with surface finishes, but none of those tested were of a resistive type. Theoretically, a wall for a house designed to prevent the occurrence of condensation should be constructed so that high resistance to vapor transmission is provided on the warm side of the wall and low resistance on the cold side. On this basis, high resistance to vapor transmission in sheathing is undesirable.

Fiberboard interior wall finish was used in a number of Government subsistence homes in a northern Wisconsin forest area, and a serious condensation problem developed during the winter of 1938-1939. Vapor resistant tests were made on the material used in this project, the test specimens painted the same as the material used in construction with a variety of additional paint coatings intended to increase the resistance. The additional paint coatings appeared greatly to increase the resistance both for side wall and ceiling material. The original paint coatings appeared to make a good base for the final coats, and it is reasonable to believe that without these original coats the final results would have been poorer.

PLASTER AND GYPSUM PRODUCTS

For new residential buildings it is preferable to use vapor barriers below or back of the plaster, but for buildings already erected, having plaster finish, some form of coating on the plaster seems to be the most practicable method of obtaining protection from exterior wall condensation. There are laths that are high in resistance, such as aluminum-foil-backed rock lath, fiberboard lath coated on the back with asphalt, wire lath backed with vapor resistive paper, etc.

Primer and sealer paints followed by finish decorative paints offer fair resistance and may offer enough resistance for those homes where the condensation problem is not serious. One coat of aluminum paint seems to offer a little better resistance than the primer and sealer, and two coats of aluminum are definitely better. The gypsum board test samples were added because of shortage of plaster samples.

MISCELLANEOUS PRODUCTS

A number of products tested did not fall into the classified lists and are tabulated separately. Some of these are standard products; some, special. Only one type of rock wool was tested, since it was presumed that it would cover other types with sufficient degree of accuracy. This type is resistive to liquid water, floats indefinitely, but allows vapor to pass through freely.

Oiled and painted cloth used as wall coverings did not show the resistance that might be expected; in fact, the tested samples fell considerably below the required resistance. Manufacturers of these products might do well to investigate the possibilities of back-up coatings or treatments that would place these materials high in the list of vapor barriers suited to houses already built.

There are a number of blanket insulations on the market, some of which have cover sheets having more or less resistance to vapor transmission, others that are covered with sheets that are merely intended to act as a support for the insulation. Some of the resistive coverings rank very high, and such blankets may be used without additional vapor barriers. Others must be supplemented with approved barriers to obtain satisfactory protection. Manufacturers are constantly working to improve the resistance of such products, and values given here may not be in keeping with the resistances of products now on the market.

PAN TESTS

The rate of evaporation from an open pan of clear water is only about 40 per cent of that from one containing sphagnum moss. A check test was made

with tin plates in place of the test sample to determine possible leakage or loss of weight in the mastic used as a sealing compound. No loss resulted. A similar test was made using the calking compound covered with foil as with the test samples but without water and here also no loss of weight was recorded.

EFFECT OF CHANGING VAPOR PRESSURE DIFFERENCES ON VAPOR TRANSMISSION

There have been a number of investigators making studies in this same field, but no common method of testing has been followed. As may be expected, the tests do not cover the same materials though one or more similar materials may appear in the lists of two investigators making it possible to compare values at least on those materials common to both lists. An effort has been made by others to bring the values obtained by different investigators into one general table by calculation. This is based on the assumption that vapor transmission is directly proportional to the differences in vapor pressure on opposite sides of the test material and that, where the vapor pressure differences are known, it is possible to determine by calculation a common vapor pressure difference that covers the results of several investigators.

Some limited investigations made in connection with the vapor transmission study, however, indicate that this method of comparison is not accurate and may be subject to serious errors. Apparently vapor transmission is not directly proportional to vapor pressure differences in all cases, particularly where materials that are hygroscopic are involved. Since many of the materials concerned in building construction are more or less hygroscopic, it would appear that there is little promise at present of comparing the results of different investigators by calculation.

After completing the regular tests in the 30 per cent relative humidity room, a number of test pans were placed in the 65 per cent relative humidity room and weighings continued. The test material represented six lots of three pans each of Douglas-fir having various coatings of aluminum paint. Both the 65 and the 30 per cent relative humidity rooms were maintained at 80 F; therefore, the exposure reduced the difference in vapor pressure on opposite sides of the test specimen by 50 per cent, and it might be expected that vapor losses would be in the same order. In the first group of 18 pans, however, it was found that the loss was reduced an average of only 27 per cent; the maximum was 42 per cent, and the minimum 12 per cent. The results are tabulated in Table 2, first cycle. Later a few more pans were available and the same tests made with similar results, which are also tabulated in Table 2, first cycle. Most of the samples were Douglas-fir plywood with various coatings, mostly aluminum, but the tests also included some fiberboard materials used for sheathing. The differences between individual test specimens in each lot of the Douglas-fir group in the 30 per cent relative humidity room were characteristic of other tests made, and each group may be regarded as reasonably consistent within itself. With the exception of two lots, the same was true in the 65 per cent relative humidity room insofar as consistency within groups was concerned.

The exceptions were sample lots 10 and 12 covered with oiled and painted cloth where the weights in the 30 per cent room never reached an equilibrium but continued to increase slightly during the period of test. This condition

TABLE 2—RELATION OF MOISTURE LOSS IN 30 PER CENT RELATIVE HUMIDITY ROOM COMPARED TO THAT IN 65 PER CENT RELATIVE HUMIDITY ROOM

Sample		Material	First cycle: Average loss in grains per square foot per hour: 30 per-cent room	First cycle: Average loss in grains per square foot per hour: 65 per-cent room	First cycle: Percent loss in 65 per-cent room compared to 30 per-cent room	Second cycle: Average loss in grains per square foot per hour: 30 per-cent room	Second cycle: Average loss in grains per square foot per hour: 65 per-cent room	Second cycle: Percent loss in 65 per-cent room compared to 30 per-cent room	Percent increase in 30 percent room second cycle compared to first cycle	Percent change in 65 percent room second cycle compared to first cycle
Number										
1-1	5/16-inch Douglas-fir plywood	One spray coat 1 pound aluminum primer one side One brush coat 1-3/4 pounds aluminum primer other side	2.505	1.452	58.0					
1-2	Do	do	2.435	1.480	60.8					
1-3	Do	do	2.540	1.770	67.0					
2-4	Do	One brush coat 1-3/4 pounds aluminum primer one side One brush coat 1-3/4 pounds aluminum primer other side	4.090	2.855	69.8					
2-5	Do	do	3.830	2.664	69.6					
2-6	Do	do	4.060	2.800	69.0					
3-7	Do	One spray coat 1 pound and one 1 brush coat 1-3/4 pounds aluminum primer one side One brush coat 1-3/4 pounds aluminum primer other side	3.230	2.270	70.3					
3-8	Do	do	3.065	2.190	71.5					
3-9	Do	do	3.410	2.440	71.6					
4-10	Do	One brush coat 1-3/4 pounds aluminum primer one side One brush coat 1-3/4 pounds aluminum primer other side	2.915	2.004	68.7					
4-11	Do	do	2.800	2.090	73.6					
4-12	Do	do	3.170	2.428	76.6					
5-13	Do	One spray coat 1 pound aluminum primer one side One brush coat 1-3/4 pounds aluminum primer other side	1.400	1.030	73.5					
5-14	Do	do	1.650	1.215	73.6					
5-15	Do	do	1.675	1.215	72.5					
6-16	Do	One spray coat 1-3/4 pounds aluminum primer "C" one side One brush coat 1-3/4 pounds aluminum primer "C" other side	1.556	1.378	88.6					
6-17	Do	do	1.690	1.480	87.5					
6-18	Do	do	1.730	1.508	87.2					
7-19	Do	One brush coat 1 pound aluminum primer "D" one side One brush coat 1 pound aluminum primer "D" other side	3.616	2.765	76.5	4.320	2.765	64.0	19.0	± 0.0
7-20	Do	do	3.562	2.675	74.8	4.160	2.810	67.5	17.0	+ 6.0
7-21	Do	do	3.990	2.930	71.0	5.210	2.965	57.0	30.0	+ .5
8-22	Do	One brush coat 1-3/4 pounds aluminum primer "D" one side One brush coat 1-3/4 pounds aluminum primer "D" other side	2.215	1.793	80.4	3.035	1.845	60.8	37.0	+ 3.0

8-23	Do.	do.	3.142	2.452	78.0	4.230	2.600	61.5	34.0	+ 6.0
8-24	Do.	do.	3.535	2.765	78.0	4.360	2.600	59.5	23.0	- 6.0
10-25	Do.	Oiled and painted cloth one side	1.768	2.030	[1]115.0	2.705	2.155	79.4	53.0	+ 5.0
10-26	Do.	do.	1.742	2.484	[1]143.0	2.800	2.200	78.5	60.0	-12.0
10-27	Do.	do.	1.768	1.622	92.0	2.135	1.805	88.7	22.0	+11.0
11-28	Do.	Oiled and painted cloth one side One brush coat 1 pound aluminum primer "D" other side	5.866	4.550	77.4	6.590	4.190	63.5	12.0	- 8.0
11-29	Do.	do.	5.620	4.222	75.0	6.240	3.870	62.0	11.0	- 9.0
11-30	Do.	do.	6.660	4.830	72.5	7.170	4.350	60.6	7.0	-10.0
12-31	Do.	Oiled and painted cloth plus 1 pound aluminum primer one side One coat 1 pound aluminum primer other side	1.262	1.500	[1]18.8	2.280	2.020	88.7	81.0	+34.0
12-32	Do.	do.	1.238	1.607	[1]30.0	2.210	1.890	85.5	79.0	+18.0
12-33	Do.	do.	1.213	1.420	[1]17.0	2.180	1.890	86.6	80.0	+33.0
12-34	3/4-inch fiberboard sheathing coated		5.920	3.830	64.6	7.380	3.430	46.3	24.0	-10.0
12-35	Do.		6.490	3.680	56.6	6.530	3.300	50.5	.5	-10.0
12-36	Do.		5.860	3.560	60.7	6.940	2.505	36.2	18.0	-30.0
12-37	Do.		2.030	1.630	80.2	3.300	2.025	62.3	62.0	+16.0
12-38	Do.		19.008	9.360	49.2					
12-39	Do.		1.132	.925	81.6	1.550	.893	57.6	37.0	- 4.0

[1] Gain

was not apparent in the weighings made on lot 11, though covered with a similar material.

The fiberboard materials, with one exception, acted about the same as the plywood. The exception was a sample having a very high rate of transmission, and in this case the rate of transmission was essentially proportional to the difference in vapor pressure.

After the first cycle of tests in the 30 and 65 per cent relative humidity rooms had been completed, the second series was run through a second cycle. This meant moving the pans from the 65 per cent room back to the 30 per cent room and later again to the 65 per cent room. The results of this test are shown in Table 2, second cycle. Without exception, the rate of loss in the 30 per cent room was higher during the second cycle than during the first. When the pans were returned to the 65 per cent room, with three exceptions the loss was consistent with the results of the first cycle through these same rooms.

There are insufficient data available to explain these inconsistencies, but it is apparent that factors, other than differences in vapor pressure, affect vapor movement through hygroscopic materials.

Evaluation of Vapor Barriers

The information available from which it is possible to evaluate the optimum resistance desired in a vapor barrier is quite limited. However, studies made over a period of years in the Laboratory test house do show: first, the importance and necessity of a vapor barrier, and second, within certain limits, the degree of resistance desired. In walls of conventional frame construction consisting of plaster, studs, sheathing and wood siding, experience has taught that moisture accumulating in the wall gathers principally in the sheathing; hence a record of the moisture content of the sheathing may be used to determine the efficiency of a moisture barrier. The test house walls were divided into panels, some with insulation, some without, some having no moisture barrier, others having moisture barriers varying in resistance. Since the temperature and humidity conditions within the test house were main-

TABLE 3—MAXIMUM MOISTURE CONTENT IN SHEATHING OF WALL PANELS IN TEST HOUSE SHOWING EFFECT OF VARIOUS TYPES OF VAPOR BARRIER

Wall panel	Inner wall	Vapor barrier	Insulation	Sheathing	Sheathing paper	Vapor transmission value in table: Lath and plaster	Vapor transmission value in table: Vapor barrier	Maximum moisture content of sheathing: Top	Maximum moisture content of sheathing: Bottom	Maximum moisture content of sheathing: Average
								Per cent	Per cent	Per cent
1-2-3	Gypsum lath and plaster	None	None	3/4-inch wood	50 pounds sheathing paper	13.850	....	16.8	23.1	20.0
4-5-6	..........Do..........	None	4-inch Rock wool	......do......	......do......	13.550	....	47.5	38.3	42.9
7-8-9	..........Do..........	50-pound coated paper	..........do..........	......do......	None	13.550	.364	[1]18.0	13.5	15.8
10-11-12	..........Do..........	Insulation back-up paper	..........do..........	......do......	Slater's felt	13.850	1.850	23.2	13.5	18.4
13-14-15	..........Do..........	50-pound coated paper	..........do..........	......do......	......do......	13.850	.364	12.4	15.3	13.8
16-17-18	..........Do..........	Foil backed lath	..........do..........	......do......	......do......	13.850	.189	15.0	14.2	14.6
19	..........Do..........	None	4-inch blown Rock wool	......do......	......do......	13.850	....	30.4	38.9	34.7
22	..........Do..........	Cover sheet	1/2-inch blanket insulation	......do......	......do......	13.550	1.400	16.0	14.9	15.4
25	..........Do..........	Cover sheet	2-inch blanket insulation	......do......	......do......	13.550	1.168	16.7	16.3	16.5
26	Fiberboard lath, gypsum plaster	None	4-inch Rock wool	3/4-inch fiberboard	......do......	14.500	....	29.0	[2]13.9	21.4
27	..........Do..........	None	None	......do......	......do......	14.800	....	18.3	[2]18.8	18.6
28	..........Do..........	None	None	3/4-inch coated fiberboard	......do......	14.800	....	25.0	16.2	20.6
29	..........Do..........	None	4-inch Rock wool	......do......	......do......	14.800	....	56.0	32.6	44.3
30	Coated fiberboard lath and gypsum plaster	Coated lath	..........do..........	......do......	......do......	....	3.240	18.2	12.6	15.4
31	Gypsum lath and paster	Slater's felt	..........do..........	3/4-inch wood	......do......	13.550	9.815	30.4	14.6	22.5
32	..........Do..........	30-30-30 Duplex	..........do..........	......do......	......do......	13.650	1.012	15.3	11.2	13.2
33	..........Do..........	30-60-30 Duplex	..........do..........	......do......	......do......	13.550	.512	13.6	10.6	12.1
34	..........Do..........	Duplex sheet, 30-90-30 reinforced	..........do..........	......do......	......do......	13.550	1.018	10.8	9.5	10.3
35	..........Do..........	Duplex sheet covered with metal oxides	..........do..........	......do......	......do......	13.550	1.080	11.1	9.6	10.4
36	..........Do..........	Foil covered kraft	..........do..........	......do......	......do......	13.850	.110	....	9.8	9.8
37	..........Do..........	One coat aluminum two coats flat paint	..........do..........	......do......	......do......	13.550	.925	14.4	11.5	13.0
38	..........Do..........	One coat primer and sealer, one coat flat paint	..........do..........	......do......	......do......	13.550	1.074	17.9	10.7	14.3
39	..........Do..........	Two coats lead and oil paint	..........do..........	......do......	......do......	13.850	1.870	21.3	12.3	16.8

[1] Average raised by one sample subject to a leak between panels. Other two samples average 15 per cent moisture content.

[2] Vapor resistance of this sheathing so low that vapor passed through it and gathered back of and in siding where it was not measured as a part of this record. Paint blisters were particularly bad on the siding over these panels.

tained constant at 72 deg and 42 per cent relative humidity, the exposure was essentially the same for all panels. The house was protected so that no variable would be introduced by exposure to the sun.

The panels are briefly described in Table 3, and the maximum moisture content attained in the sheathing bears a more or less definite relation to the resistance to vapor transmission of the vapor barrier. The moisture content of the sheathing was determined by removing previously prepared sections or samples of the sheathing, weighing them, and replacing them in the wall. The weights included any ice present on the inner face of the section at the time of inspection. Generally, it was found that as the ice melted the water soaked into the sheathing with very little change of weight. Before the first weights were taken in the fall the samples were placed in a chamber maintained at constant temperature and humidity and brought to a constant weight or common moisture content of 6 per cent. When replaced in the wall they picked up moisture until they had about 10 per cent, which seemed to be the equilibrium moisture content for all panels at that time of year. Sometime later the outside temperature dropped to an average of about 35 deg, and at once the unprotected samples picked up moisture rapidly; the protected samples, slowly. The moisture content continued to pick up in all samples as the weather continued to get colder. The protected samples practically reached their maximum content early in January, and there was very little change during a severe cold weather period that followed late in January and in February. Apparently the inflow of vapor exceeds the outflow until some vapor pressure head is established, after which the outflow equals the inflow until, when outdoor temperatures average 40 deg or higher, the outflow exceeds the inflow and the moisture content tends to drop until it reaches about 10 per cent. The maximum moisture content reached appears to be related to the degree of resistance of the vapor barrier, and is lowest in the more resistive barriers.

From the information at hand, it appears that for a satisfactory vapor barrier the maximum allowable transmission value for any individual piece should not exceed one grain per hour per square foot, on a basis of the vapor pressure difference used in the foregoing tests. Since individual pieces vary in vapor transmission, the average value for a test lot of samples would be less than one grain per hour. Specifications for acceptable vapor barriers might be expressed in either of two ways: (1) the maximum allowable transmission rate per square foot per hour for any individual sample of a test lot, or (2) an average value that would be low enough to assure that individual pieces would fall below the maximum allowable rate of transmission. For (1) the maximum transmission value could be established as one grain per hour per square foot and (2) an average of 0.600 grain per hour would be reasonable assurance that individual pieces would fall within acceptable ranges. Test material should be representative of the product as prepared for shipment. For example, types of insulation having attached vapor barrier should have test samples stripped from the insulation after the product has been prepared for shipment. Blanket types that are shipped in rolls should have samples taken from near the center of the roll. Other types of rolled material should have samples taken from the center of the roll or where the sharpest bends occur.

The unprotected samples did not show a tendency to equalize but continued to increase in moisture content as long as the temperature was below freezing. Late in March there was a sharp rise to above 40 deg when they started

to lose moisture. This loss continued as the outdoor temperatures continued to rise during April and May. It was nearly the middle of June, however, before the samples having the highest winter moisture content reached about 10 per cent, which appeared to be the summer equilibrium condition.

For comparative purposes, 15 per cent has been arbitrarily selected as the maximum moisture content for sheathing, and vapor barriers that can consistently hold the moisture content at or below this value may be considered satisfactory. This moisture content has been chosen as representing the condition of well air-dried material, and limits the seasonal moisture change to about 5 per cent. (The summer minimum is about 10 per cent moisture content, as explained previously.) Moreover, it allows a factor of safety to take care of minor leakage and breaks in the barrier.

It appears that the barriers having a resistance of about one grain per hour will hold the moisture content of the sheathing at or below 15 per cent moisture content. Some of the barriers in this class, for example the 30/30/30 duplex, represent materials that vary widely in resistance. Specifically it would seem acceptable to admit a material having a known average resistance equal to one grain per hour on a basis of the laboratory method of test. This would admit a 30/30/30 grade duplex of one company and exclude that of another.

Paint coatings on plaster may be very effective as moisture barriers if materials are properly chosen and applied. Of these, aluminum primers, one or two coats, followed by two decorative coats of flat paint or lead and oil seem to offer the best resistance. Primer and sealer paint on the plaster followed by decorative finish coats offer fair resistance and may be satisfactory in older houses though the resistance is below the standard desired in new houses. The plaster finish on the test walls and test samples was smooth. A rough or sand finish plaster might not have so high a vapor resistance with the same types of paint coatings as did the smooth finish.

GENERAL CONCLUSIONS

The importance of vapor barriers in reducing moisture accumulation in walls is shown in Table 3. Even the unprotected and uninsulated wall panels 1-2-3 need protection; but when insulated and unprotected, as in panels 4-5-6, the moisture content of the sheathing reaches a very high point. The same condition appears in the walls having fiberboard insulation. From results in the test walls and in field observations, it is apparent that walls having fiberboard sheathing, whether coated or uncoated, should be protected by vapor barriers.

A variety of suitable materials is available for use as vapor barriers. For buildings under construction the barrier may be one of the asphalt-coated papers, duplex papers, foil-backed lath, suitably coated fiberboard lath, or a high resistance back-up paper on the insulation. For buildings after completion, the barriers are generally limited to surface paint coatings.

For dry wall construction, where plywood, fiberboard, or other wall materials are used in place of plaster, paint coatings may be applied to the back or to the face. Asphalt paint coatings on the back of plywood, for example, make an excellent vapor barrier. Two coats, or enough to make a bright shiny surface, are required.

Since vapor barriers are not 100 per cent resistant to vapor movement, exterior wall materials should be permeable to vapor. This condition exists with all usual side-wall materials, such as wood, brick, stone, concrete, stucco,

and the like. Any sheathing paper used should be permeable to vapor and resistant to water. This requirement is met with slaters' felt or tar paper. On the other hand, asphalt-saturated felts are several times as resistive as the tar felts and should not be used as a sheathing paper where material of low vapor resistance is required. Many roofing materials are excellent vapor barriers, particularly asphalt shingles, roll roofing, tin roofs, and composition roofs. To eliminate any moisture that works through the ceiling vapor barrier into the attic or roof space, some form of ventilation should be provided in the attic or roof space above the insulation to allow the escape of the moisture. This ventilation should be from the outside, and there should be no free openings into the attic or roof space from the heated portions of the house during cold weather.

DISCUSSION

E. C. LLOYD, Lancaster, Pa. (WRITTEN): The long and thoroughly prepared background of Mr. Teesdale and the Forest Products Laboratory on vapor flow work is evident in this paper which constitutes a valuable contribution to the subject of vapor flow through building materials.

It seems to me that one point made by the author is worthy of particular emphasis. I refer to the statement "Apparently vapor transmission is not directly proportional to vapor pressure differences in all cases, particularly where materials that are hygroscopic are involved." Actually, the data in the paper as presented in Tables 1 and 2 are all presented in support of the first clause of the quoted statement, but to me there seems to be little that bears out the latter clause, though it is reasonable to suppose that the hygroscopicity of the specimen under test has a decided bearing on the rate of vapor flow. The particular significance to the writer is twofold—*first,* where experimental results are expressed on a unit basis; *i.e.,* in grains per square foot, per hour, per inch of mercury vapor pressure, care should be taken to record the high and low vapor pressures and not alone the differential pressure and, *second,* where experimental results are expressed on a *condition of test* basis, as is the case here, record should be similarly shown of the high, low and differential vapor pressures prevailing in the test.

The author has given a figure of 0.600 grains per square foot per hour for the conditions of the test as representing the maximum vapor transmission allowable in a suitable vapor barrier. This might be expressed as 0.830 grains per square foot, per hour, per inch of mercury vapor pressure where the conditions prevailing are a temperature of 80 F on both sides of the specimen with saturated atmosphere on the one side and 30 per cent humidity on the other.

Referring to Table 2, it would seem that none of the samples tested showed either in the 30 per cent or 65 per cent room a resistance to vapor flow which satisfactorily meets the condition for a suitable vapor barrier of 0.600 grains per square foot per hour. This is rather surprising in view of the fact that relatively impermeable surface finishes were in place on one or both sides, and, further, the plywood had the several glue lines to supplement the outside seal. If possible, I should be interested to know the number of pieces and the type of glue used in the 5/16 in. plywood of Table 2, and, also, I should be interested to have a little further data on the six fiberboard sheathings reported on in the same table. At least, with respect to the sheathings, I would be interested to know whether the six samples represent one product as made by one manufacturer and if they are simply a fiberboard with a dipped or roller-coated asphalt surface on each face.

On the data in Table 3, a number of questions arise. I would be interested in knowing if the vapor transmission values for the lath and plaster as recorded in the seventh column were determined, as with the vapor barriers, by a humid pan method in the 80 F—30 per cent room or if they were determined by actual exposure which would involve a temperature gradient as well as a vapor pressure differential.

H. E. LEWIS, Toledo, Ohio: These remarks are not in criticism but merely a request for additional work; if not research work, a decision on phases of vapor transmission through different types of building materials.

This paper gives some very helpful data if the individual papers or woods could be identified. The only recourse at present is to try to decide which one of the papers listed anonymously in this and other research papers is the one that you have in mind.

Referring to other reports of a similar nature, such as the one in *Heating and Ventilating* for September, 1942, the individual products are not identified by manufacturer's name.

If authors of vapor transmission studies would get together on a common test method for vapor transmission, plus a common set of units for specifying transmission, the results would be much more helpful in the field.

As to units, we have vapor transmission expressed in grams per square foot per hour per inch of mercury. Or we might have it in grams or grains per 100 sq in. per pound per square inch vapor pressure differential, or other units.

The statement by the author that vapor transmission is not proportional to the vapor pressure differential must be clarified from a technical standpoint with other authorities.

We should have information available on vapor transmission, just as we have on heat transmission, standard test procedure and definite values for different materials, identified by group, by weight, or other physical characteristics for reference use.

If such information already is available, I certainly would appreciate knowing where to find it. I have tried for a number of years, but never have been able to put my finger on the precise data that are required for estimating work in the vapor transmission field. That is about all I have to say, except that I hope this will raise some discussion by others interested in vapor transmission from an application standpoint.

J. N. HADJISKY, Birmingham, Mich.: In 1931 and '32 I had a problem in which certain materials had to be stored in a room 8 ft high, 14 ft wide and 30 ft long, to be kept at 85 to 90 F temperature, and, if possible, 30 per cent relative humidity. We selected a dehumidifying unit and it was found that it was just 100 per cent too small. The construction of the room was 4 in. of cork with 1 in. ship lap board on each side. We ordered the inside painted with three coats of shellac and the floor painted with two coats of paint. The floor was concrete. We still continued to get an unexplainable amount of moisture, which I assumed came by diffusion through the windows and door openings.

In order to prove my contention I constructed a small chamber of glass, left a few openings that would allow diffusion, and to my surprise, I found a considerable amount of moisture entering through the very limited apertures.

Some three or four years later something similar came to my attention, a specially constructed double glass insulation 10 ft square, in a swimming pool, in which the space was about 1¼ in. between. There was an enormous amount of condensation considering the amount of diffusion and the number of openings.

I am convinced that probably in such matters we ought to look into the partial pressure affecting diffusion, diffusion of light gases, steam as against air, and inasmuch as the preamble of the paper states that it is the condensation in houses we are interested in, we ought to look into the question of the cracks around the windows where the moisture comes mostly through vapor partial pressure diffusion rather than through the surfaces.

MR. LEWIS: I was not thinking particularly in terms of home insulation applications but vapor transmission in general; through all types of constructions, such as low temperature equipment or refrigerated piping. We have the same problem involved there that we have in vapor barriers for home construction.

Perhaps it is not within the scope of the A.S.H.V.E. to consider all applications, but insofar as home construction materials are concerned, the pattern might be established for vapor trañsmission in general.

A. S. BULL, Minneapolis, Minn.: The work in vapor transmission has been of great interest to insulation board people. We were sorry to find that the work at the Forest Products Laboratory was curtailed due to the emergency of the war. I believe that the only laboratory that is continuing work on that subject at the present time is at the University of Minnesota.

In regard to the point raised by Mr. Lloyd, the possibility of calculating vapor transmission and the fact that the author indicates that it is of doubtful possibilities, recent work indicates that it may be done for practical purposes.

One of the other speakers has indicated his interest in the practical aspects of this problem, and I have recently seen calculations made indicating that for practical purposes you can get close approximations; in other words, the author's valuable test data and those from other investigators do not necessarily have to remain in tables only for laboratory use. I think there are possibilities that that test data can be translated into practical available field data for field use.

Finally, regarding the use of a standard method of stating results, the author has used several methods of giving results, either in grains or grams and so on. The method used in the current paper, of grains per square foot per hour is an advance, because, certainly, using grains is an English system. If you mix grams with square feet you are mixing the metric system and the English system.

About a year ago the *Insulation Board Institute* discussed the matter of standard units and made a suggestion which was published in one of the magazines that a standard method of stating results could be grains per square foot per hour per inch of mercury. Now, the only thing that does not check between that and the author's units is that he does not go to his final unit of vapor pressure difference.

From the practical side we would all agree that a uniform method of stating results is desirable, and, unless some other, better method is adopted, I think the proposal submitted by P. D. Close a year ago in the publication of grains per square foot per hour per inch of mercury is one that merits consideration.

AUTHOR'S CLOSURE: In answer to Mr. Lewis' comment on my statement that vapor transmission is not proportional to the vapor pressure differential, this is drawn from experience and work other than that cited in the present report and the results given in Table 2 are merely shown to indicate that the vapor transmission is not always changed in direct proportion to the change in vapor pressure. In the 80 F-30 per cent relative humidity room the vapor pressure inside the pan was 1.03 in. Hg and outside 0.31 in. Hg. In the 80 F-65 per cent relative humidity room the vapor pressure inside the pan was 1.03 in. Hg as before and outside 0.67 in. Hg.

It seems desirable that an investigation should be carried out to obtain additional data regarding the effect of vapor pressure difference upon vapor transmission through various types of materials, and it should include variations in vapor pressure due not only to changes in relative humidity but also to changes in temperature. If vapor pressure differences are not in line with vapor transmission figures, then the only solution of the problem is to adopt a fixed test procedure to be used in future research.

One coat of aluminum paint is not a highly impermeable coating and unless two or three coats were applied on one face we would not expect exceptional resistance to moisture movement. One coat of aluminum paint applied to opposite faces of a barrier would not have the same resistance as if two coats were applied to one side. The effectiveness of one, two, and three coats of several materials is shown in a bulletin.[3] The 5/16 in. plywood was 3-ply, bonded with soy bean glue.

The first three 3/4 in. fiberboard samples were made by the same manufacturer and the last three came from three different sources. At present no information is available concerning the method of coating.

The values for lath and plaster were determined in an 80 F-30 per cent relative humidity room on one side and a saturated atmosphere on the other. No temperature gradients were involved.

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[3] Coatings for Minimizing Changes in the Moisture Content of Wood (*Technical Note No. 181*, Forest Products Laboratory).

No. 1231

SUMMER COMFORT FACTORS AS INFLUENCED BY THE THERMAL PROPERTIES OF BUILDING MATERIALS †

By C. O. Mackey * and L. T. Wright, Jr.,** Ithaca, N. Y.

Purpose

The purpose of this study is to analyze the effects of the thermal properties of wall materials influencing human comfort under typical summer conditions. This study is restricted to the case of a wall of a single layer of homogeneous material. With *steady heat flow* through a building wall, the amount of heat entering one surface of the wall in unit time is the same as that leaving an equal area of the opposite surface; there is no storage of heat in the building material. As far as properties of the material are concerned, two walls have identical steady heat flow characteristics when they have the same absorptivity for solar radiation of the outside surface, the same ratio of thickness to thermal conductivity, and the same emissivity of the inside surface.

Steady flow of heat through building walls is the exception rather than the rule. If the wall has any heat storage capacity, and all wall materials in any finite thickness do, then steady heat flow can exist only when (*a*) the temperature of the indoor air and the motion of that air remain constant with respect to time, (*b*) the amount of radiation received or emitted by the inside surface of the wall remains constant, (*c*) the temperature and motion of the outdoor air remain constant, and (*d*) amount of radiation received or emitted by the outside surface of the wall remains constant. These conditions are seldom, if ever, exactly constant through a one-hour period and are never constant throughout the day. It is basically incorrect to select a material for a building wall on the basis of its steady flow thermal properties; engineers are aware of this fallacy, but the laws of unsteady heat flow are very complex and departures from the steady flow laws are not easy to derive in any case and may not be important in some cases.

The *unsteady flow* thermal properties of the building wall will be more important in the summer cooling season than during the winter heating season. The daily range in the outdoor air temperature is commonly greater in the summer than in the winter months. Furthermore, the difference between the temperature of the indoor and outdoor air is much greater in winter than in summer; hence, the ratio of the daily range in outdoor air temperature to temperature difference is far greater in summer than in winter. Similarly,

† This paper is the result of research sponsored by the John B. Pierce Foundation in its program in the field of low-cost housing and a report of complete study will be published by the Foundation.
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Presented at the 49th Annual Meeting of the American Society of Heating & Ventilating Engineers, Cincinnati, Ohio, January, 1943.

the daily range in the solar energy incident upon a given building wall has a greater proportionate effect in summer than in winter. Except for the case of intermittent heating, it is logical to judge the thermal effectiveness of a building material for the heating season on the basis of the steady flow thermal properties, but for the cooling season, such an analysis is not adequate.

The factors involved in this study may be briefly summarized. The feeling of warmth or cold experienced by human occupants of an enclosure is influenced by (*a*) the temperature of the indoor air, (*b*) the motion and relative humidity of the indoor air, (*c*) the temperature and absorptivities (emissivities) of solid surfaces within the enclosure *seen* (in a radiant sense) by the occupants; also the angle factors throughout which the occupants *see* these solid surfaces. The effects of air motion and humidity are not considered further, because any such effects are independent of the thermal properties of the wall materials. The higher the indoor air temperature and the higher the temperature of the solid surfaces *seen* by the occupant, the warmer will that occupant feel, as indicated by the exhaustive studies carried on at the John B. Pierce Laboratory of Hygiene, New Haven, Conn. The air-conditioning engineer generally thinks of the temperature, humidity, and motion of the indoor air when he thinks of human comfort, for these properties are under his control. He takes what he gets in temperatures of inside building surfaces, for he has not selected the building materials or planned the orientation and room usage. The temperatures of these solid surfaces are important in fixing the rate of heat exchange with the human body by radiation, and they cannot be ignored in any complete study of comfort factors.

The hour-by-hour variation of the temperature of the inside surface of a wall panel will depend upon (*a*) the hour-by-hour variation of the outdoor air temperature; (*b*) the hour-by-hour variation of the solar radiation incident upon the outside surface of the wall; (*c*) the motion of the outdoor air over the wall; (*d*) the absorptivity for solar radiation and the shading of the outside surface of the wall; (*e*) the thickness, thermal conductivity, and volumetric specific heat of the wall panel; (*f*) the hour-by-hour variation of the temperature of the indoor air; (*g*) the emissivity (or absorptivity) for low temperature radiation of the inside surface of the wall; (*h*) the motion of the indoor air over the wall; (*i*) the temperatures and emissivities (or absorptivities) for low temperature radiation of all solid surfaces within the room with which the inside surface of the wall panel may be exchanging heat by radiation; also the solid angles throughout which the wall panel *sees* these surfaces; and (*j*) the temperature and rate of flow of any air supplied and exhausted to produce an air change in the enclosure.

In this list of factors affecting wall temperatures, and therefore comfort, even when no cooling, dehumidifying, or ventilating equipment is installed, factors (*d*), (*e*), (*g*), and (*i*) are under the control, in whole or in part, of the achitect and builder.

A complete mathematical solution for the hour-by-hour variation of the temperature of each inside solid surface of a building involves setting up a heat balance for each such surface, and because of the large number of variables, is impractical, particularly when unsteady heat flow is involved. Instead, certain simplifying assumptions are made in this study—assumptions which will not limit the investigation of the general effects of the thermal properties of the wall materials upon comfort.

METHOD AND RESULTS

For the *steady flow* of heat through a building wall, the temperature of the inside surface of the wall is

$$t_o = t_i + \frac{U}{h_o}(t_e - t_i) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

t_o = the temperature of the inside surface of the wall, F;
t_i = the dry-bulb temperature of the indoor air, F;
U = the over-all coefficient of heat transfer for steady flow, Btu/(hr) (ft²) (F);

$$= \frac{1}{\frac{1}{h_o} + \frac{1}{h_L} + \frac{L}{k}}$$

h_o = equivalent inside air film coefficient of heat transfer, Btu/(hr) (ft²) (F);
h_L = the outdoor air film coefficient of heat transfer, Btu/(hr) (ft²) (F);
L = the thickness of the wall, ft;
k = the thermal conductivity of the wall material, Btu/(hr) (ft) (F);
t_e = the equivalent temperature of the outdoor air to include solar radiation incident upon wall, F;

$$= t_a + \frac{aI}{h_L};$$

t_a = the dry-bulb temperature of the outdoor air, F;
I = the intensity of solar radiation incident upon the outside surface of the wall, Btu/(hr) (ft²);
a = the absorptivity of the outside wall surface for solar radiation.

Of these terms, all are familiar except the equivalent outdoor air temperature. This is the temperature of the outdoor air which, in contact with a shaded wall, would give the same rate of heat transfer and the same temperature distribution through the wall as exists with the actual outdoor air temperature and incident solar radiation. For example, when the temperature of the outdoor air is 90 F, and the intensity of solar radiation incident upon a wall with an absorptivity of 0.4 is 100 Btu/(hr) (ft²), the equivalent temperature of the outdoor air is

$$\left[90 + \frac{0.4(100)}{4}\right]$$

or 100 F for an outdoor air film coefficient of heat transfer of 4 Btu/(hr) (ft²) (F); in other words, an air temperature of 100 F in contact with a shaded wall is equivalent to the combined effects of actual air temperature and solar radiation.

For the *unsteady flow of heat* through a building wall of a single homogeneous material *with a constant temperature of the indoor air,* the solution presented by Alford, Ryan, and Urban[1] has been checked by the authors and is given here in slightly different form.

$$t_a = t_m + t_o \cos(\omega_o\theta - a_o) + t_1 \cos(\omega_1\theta - a_1) + \quad . \quad . \quad . \quad . \quad (2)$$

[1] Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939.)

Assume a Fourier series for the solar radiation incident upon the wall of the form:

$$I = I_m + I_o \cos(\omega_o\Theta - \beta_o) + I_1(\omega_1\Theta - \beta_1) + \quad \ldots \ldots \quad (3)$$

The temperature of the inside surface of the wall is

$$t_o = t_i + \frac{\frac{1}{h_o}(t_m - t_i) + \frac{aI_m}{h_o h_L}}{\frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_L}} + \sum \lambda_n t_n \cos(\omega_n\Theta - \alpha_n - \phi_n)$$

$$+ \sum \frac{\lambda_n a I_n}{h_L} \cos(\omega_n\Theta - \beta_n - \phi_n) \quad \ldots \ldots \quad (4)$$

where

$$\lambda_n = \frac{1.414 h_L k \sigma_n}{\sqrt{f_n^2 + g_n^2}} \quad \ldots \ldots \quad (5)$$

$$\phi_n = \tan^{-1}\left(\frac{f_n - g_n}{f_n + g_n}\right) \quad \ldots \ldots \quad (6)$$

$$\sigma_n = \sqrt{\frac{\omega_n \rho c}{2k}} \quad \ldots \ldots \quad (7)$$

$$\begin{aligned} f_n = {} & (h_o + h_L) k\sigma_n (\cos\sigma_n L \cosh\sigma_n L + \sin\sigma_n L \sinh\sigma_n L) \\ & + h_o h_L \sin\sigma_n L \cosh\sigma_n L + 2k^2\sigma_n^2 \cos\sigma_n L \sinh\sigma_n L \quad \ldots \quad (8) \end{aligned}$$

$$\begin{aligned} g_n = {} & (h_0 + h_L) k\sigma_n (\cos\sigma_n L \cosh\sigma_n L - \sin\sigma_n L \sinh\sigma_n L) \\ & + h_o h_L \cos\sigma_n L \sinh\sigma_n L - 2k^2\sigma_n^2 \sin\sigma_n L \cosh\sigma_n L) \quad \ldots \quad (9) \end{aligned}$$

The solution of these equations is involved and a detailed discussion is valueless at this point. The following general conclusions may have some practical value:

1. When the outdoor air temperature and solar radiation incident upon a wall are cyclic with a period of 24 hours, and when cooling equipment is operated within the structure, the total gain of heat through the wall in 24 hours depends only upon the steady flow thermal properties of the wall. All walls with the same ratio of thickness to thermal conductivity, same absorptivity of the outside surface for solar radiation, and the same emissivity of the interior surface will transmit the same quantity of heat daily. The quantity of heat transmitted daily is

$$q = \frac{24\left(t_m + \frac{aI_m}{h_L} - t_i\right)}{\frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_L}} \quad \ldots \ldots \quad (10)$$

When no cooling equipment is operated within the enclosure, the net daily gain of heat is zero. In this case, the assumed constant temperature of the indoor air must be

$$t_i = t_m + \frac{aI_m}{h_L} \quad \ldots \ldots \quad (11)$$

2. The temperatures of the inside surfaces of the wall will follow markedly different curves, when plotted against time of day, for walls having the same steady flow thermal properties but different thermal conductivities and volumetric specific heats.

In comparing two walls of the same steady flow thermal properties but different thermal conductivities and specific heats, the temperature of the interior surface of one wall will be *higher* than that of the other during certain hours of the day but *lower* during the remainder of the day. In other words, it is impossible for one

wall which has the same steady flow thermal properties as a second wall to have a lower inside surface temperature than the second wall during the entire 24-hour period.

This result affects the required capacity of the cooling equipment when cooling is used and affects the comfort at a given time of the day when no cooling is provided.

3. The selection of the thickness and thermal properties of the ideal wall is intimately tied to the use of a room at any particular time of day. A room used during the night should have walls of different properties than a room used only during the daylight hours. The practice of building a structure with all walls of the same thickness and thermal properties cannot be justified from thermal considerations alone.

In support of these general conclusions, the authors present a few results from an extended study made for the John B. Pierce Foundation. Details of

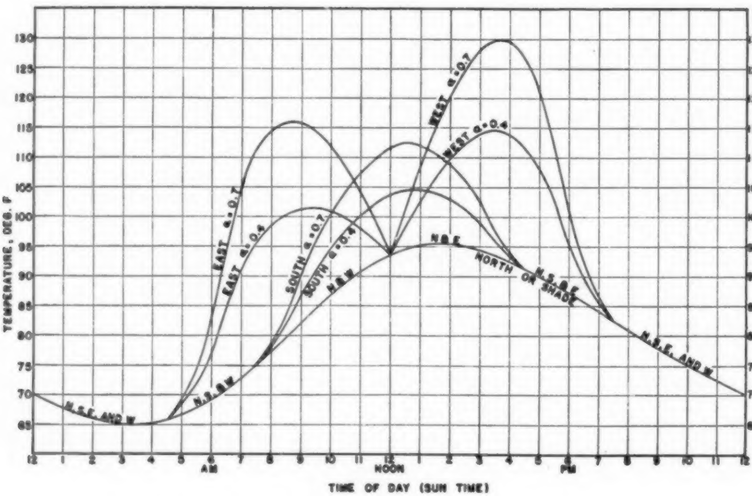


FIG. 1. EQUIVALENT OUTDOOR AIR TEMPERATURES FOR FIGS. 2 AND 3

calculation will be found in Appendix A. In Fig. 1 are shown equivalent outdoor air temperatures for the variations of outdoor air temperature and incident solar radiation assumed in this study for North (or shade), East, South, and West walls; three values of absorptivity for solar radiation are included: 0, 0.4, and 0.7. The equivalent temperature curve for zero absorptivity is the curve of outdoor air temperature. In Fig. 2 are shown the hourly variations in the temperatures of the inside surfaces of two walls with substantially the same steady flow thermal properties when exposed to the air and solar conditions of Fig. 1. The thicknesses and thermal properties of the two materials so compared are given in Table 1.

Additional assumptions made in obtaining these results are:

Outdoor air film coefficient of heat transfer = h_L = 4 Btu/(hr) (ft^2) (F);
Solar absorptivity of each exterior surface = a = 0.7;

Constant indoor air temperature = t_1 = 80 F;

Rate of heat transfer at inside surface = h_0 = 1.5 Btu/(hr) (ft^2) for each degree of difference between the temperature of the inside surface of the wall and 80 F.

A study of these curves shows that the temperature of the inside surface of the cellular glass wall follows a curve similar in shape to that of the equivalent outdoor air temperature (Fig. 1) for any given orientation; due to the low heat storage capacity of this wall, the results are substantially the same as for steady heat flow. The high heat storage capacity of the thick brick wall produces a large time lag and a considerable attenuation of the thermal effects. Although for a given orientation both walls transmit the same net quantity of heat daily (proportional to the area between the curve and the 80 F line), the

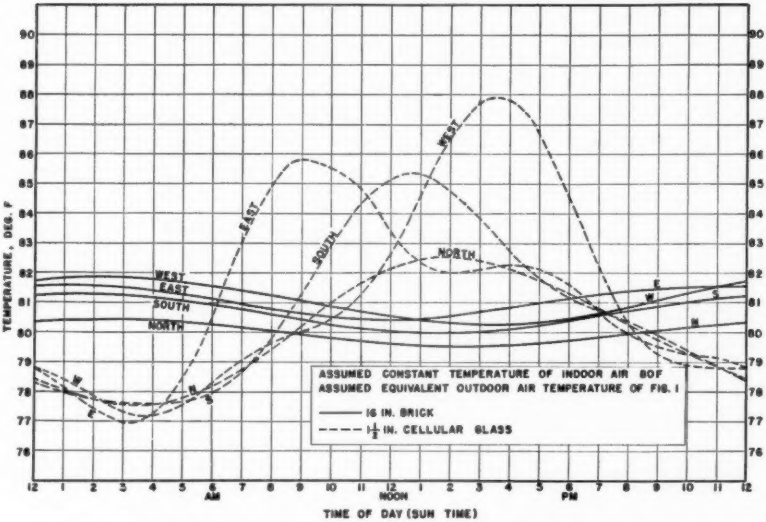


FIG. 2. TEMPERATURE OF INSIDE SURFACE OF WALL

instantaneous maximum rate of heat transfer from the inside surface of a West wall, for example, is 11.85 Btu/(hr)(ft^2) around 3:45 p.m. for the cellular glass and only 2.79 Btu/(hr)(ft^2) around 2 a.m. for the brick. For the West wall, the minimum rate of heat transfer from the inside surface of the brick wall is 0.42 Btu/(hr)(ft^2) at 3:30 p.m.; for cellular glass West wall at 3:45 a.m. there is a rate of heat transfer *to* the inside wall surface of 4.26 Btu/(hr)(ft^2), which means that *heating*, instead of cooling, would be required during the evening to maintain the assumed constant indoor air temperature of 80 F with this wall. Note also that the cellular glass West wall, for example, gives a *cooler* comfort condition than the brick wall between the hours of 7:45 p.m. and 10:30 a.m. and a *warmer* comfort condition between the hours of 10:30 a.m. and 7:45 p.m., even though the temperature of the indoor air is held constant at 80 F.

For an enclosure which is not air-conditioned, it is difficult to derive the

complete particulars of the effects of thermal properties of the wall upon comfort from a mathematical solution of a case where the temperature of the indoor air is assumed to remain constant. In a report submitted to the John B. Pierce Foundation, the authors have presented a solution which gives the hourly variation of the temperature of the inside surface of the wall and of the indoor air for a fixed, constant rate of ventilation of the enclosure with outdoor air. The results of this study of unsteady heat flow are too complex for inclusion in this paper. Instead, some general conclusions may be drawn by giving here the results obtained when the indoor air temperature is assumed to remain constant at the mean outdoor equivalent air temperature given by Equation 11. For the different orientations with $a = 0.7$ and $h_L = 4$, these mean temperatures are:

North or shade wall:	$t_i = 80$ F
East and West walls:	$t_i = 86.58$ F
South wall:	$t_i = 84.04$ F

Based upon these assumed values of a constant indoor air temperature, the hourly variations of the temperatures of the inside surfaces of the walls of

TABLE 1—PROPERTIES OF THE WALLS COMPARED IN FIG. 2

MATERIAL	THICKNESS L FT	THERMAL CONDUCTIVITY k BTU/(HR)(FT)(F)	THERMAL CONDUCTANCE $\frac{k}{L}$ BTU/(HR)(FT^2)(F)	VOLUMETRIC SPECIFIC HEAT ρc BTU/(FT^3)(F)
Cellular Glass........	0.125 (1.5 in.)	0.04	0.32	1.71
Brick (low-density)...	1.333 (16 in.)	0.417	0.313	19.9

brick and of cellular glass are shown in Fig. 3. The results for a North wall are not included, since they are the same as those shown in Fig. 2. In the case of the non-cooled enclosure, the temperature of the indoor air is not constant, but there are many solid surfaces within the room which remain at nearly these mean temperatures throughout the day, and the inside wall surface may lose heat when its temperature is above this mean and gain heat when its temperature is lower than the mean. As a consequence, the results obtained are not wholly fictional.

Realizing that the engineer and architect may gain little useful information from inspecting a solution in the form of Equations 2 through 9, or even from a study of curves for a few specific cases, the next purpose of the paper is to give a simplified solution for this case of unsteady heat flow and to present the final results in the form of a chart which gives a real picture of the effects of thermal properties upon comfort. From a study of many results, it has been found that a reasonably accurate curve of the hourly variation in temperature of the inside surface of a single-layer, homogeneous wall may be quickly found. *First,* draw a curve of *equivalent outdoor air temperature,* like Fig. 1; if the curves assumed here for outdoor air temperature and incident solar radiation

do not fit the problem, different ones may be used. *Second,* calculate the *fundamental* time lag (neglect harmonics). For this purpose, use

$$\sigma_o = \sqrt{\frac{0.1309\ \rho c}{k}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Then the fundamental time lag in hours is ϕ_o (the fundamental lag angle from Equation 6) divided by 15. *Third,* calculate the fundamental equivalent thermal resistance ratio, λ_o, from Equation 5. Then, begin with any temperature (t_{e1}) on the curve of equivalent outdoor air temperature at the time θ_1. Add the

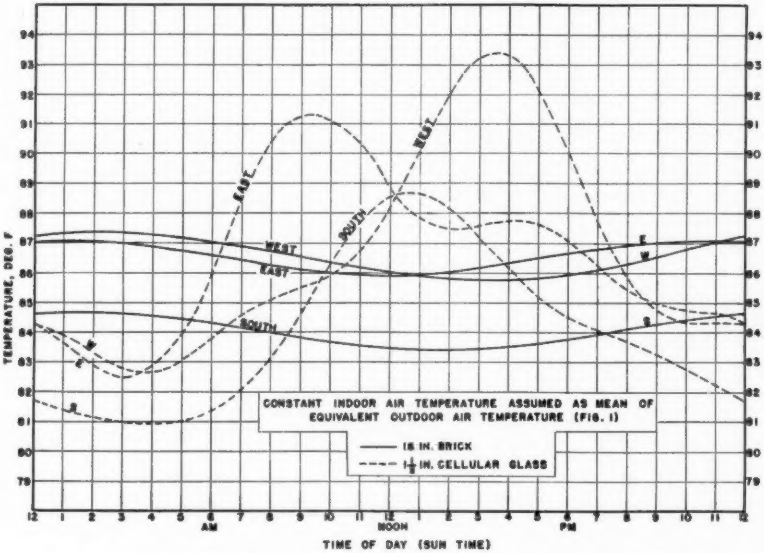


FIG. 3. TEMPERATURE OF INSIDE SURFACE OF WALL

time lag to the hour θ_1 and find the temperature of the inside surface of the wall at this new time from the steady-flow Equation 1, using λ_o in place of $\frac{U}{h_o}$; or at the time $(\theta_1 + \frac{\phi_o}{15})$ the temperature of the inside surface is

$$t_o = t_i + \lambda_o (t_{e1} - t_i) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

An example will illustrate the procedure. Consider the 16 in. brick West wall, for which results have previously been given. For this wall, $a = 0.7$, $k = 0.417$, $\rho c = 19.9$, and $L = 1.333$. The fundamental lag angle will be found to be 180 deg (time lag of 12 hours), and the fundamental equivalent thermal

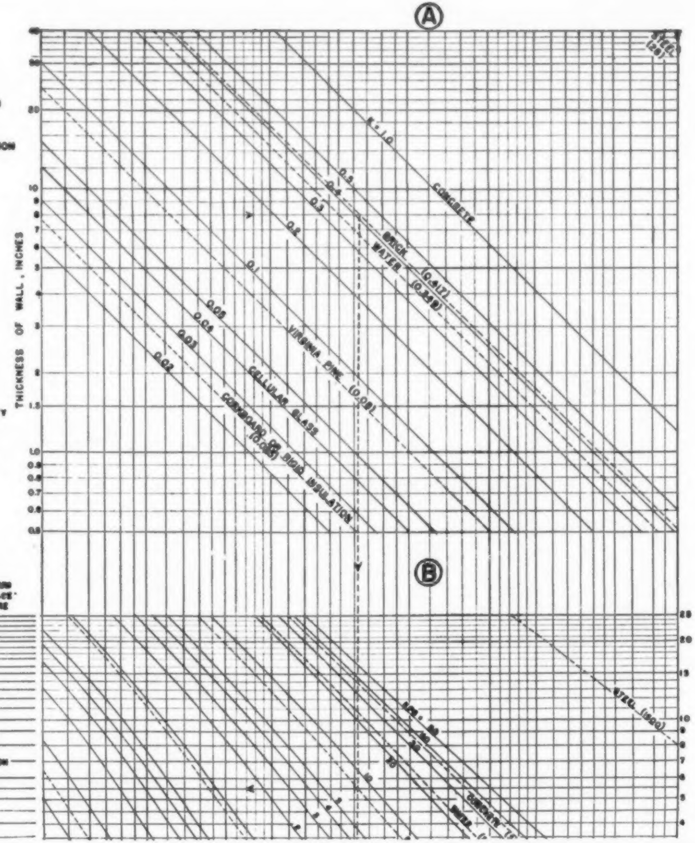
MAXIMUM AND MINIMUM TEMPERATURES
OF THE INSIDE SURFACES OF SUMMER WALLS AND THE TIMES OF OCCURRENCE OF THESE TEMPERATURES

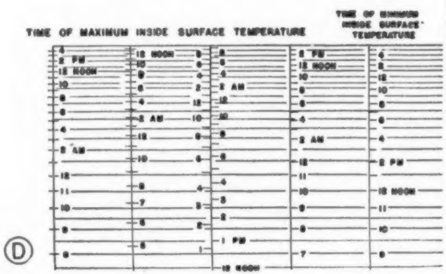
K = THERMAL CONDUCTIVITY OF WALL MATERIAL, BTU/(HR)(FT)(F)
ϱ = DENSITY OF WALL MATERIAL, LB/FT³
c = SPECIFIC HEAT OF WALL MATERIAL, BTU/(LB)(F)
a = ABSORPTIVITY OF EXTERIOR SURFACE FOR SOLAR RADIATION

ASSUMPTIONS:
CONSTANT INDOOR AIR TEMPERATURE
OUTDOOR AIR FILM COEFFICIENT OF HEAT TRANSFER OF 4 BTU/(HR)(FT²)(F)
TYPICAL HOURLY VARIATION IN TEMPERATURE OF OUTDOOR AIR
TYPICAL HOURLY VARIATION IN INCIDENT SOLAR RADIATION
RATE OF HEAT TRANSFER AT INSIDE WALL SURFACE OF 1.5 BTU/(HR)(FT²) FOR EACH DEG OF TEMPERATURE DIFFERENCE BETWEEN SURFACE AND INDOOR AIR

INSTRUCTIONS FOR USE:
1. ENTER CHART A WITH WALL THICKNESS; GO HORIZONTALLY TO RIGHT TO THERMAL CONDUCTIVITY, K, (OR MATERIAL NAME)
2. GO VERTICALLY DOWN TO PRODUCT OF CONDUCTIVITY AND VOLUMETRIC SPECIFIC HEAT, Kϱc, (OR MATERIAL NAME) ON CHART B
3. GO HORIZONTALLY TO LEFT TO PROPER SCALE INCLUDING EFFECT OF ABSORPTIVITY AND ORIENTATION; READ TIMES OF DAY OF MAXIMUM AND MINIMUM TEMPERATURES
4. CONTINUE VERTICALLY DOWN ON 2 TO PRODUCT OF CONDUCTIVITY AND VOLUMETRIC SPECIFIC HEAT, Kϱc, (OR MATERIAL NAME) ON CHART C
5. GO HORIZONTALLY TO LEFT TO PROPER SCALE AND READ MAXIMUM AND MINIMUM TEMPERATURES OF INSIDE SURFACE

EXAMPLE
8 IN BRICK WALL FACING EAST; K = 0.417; Kϱc = 8.3; a = 0.7





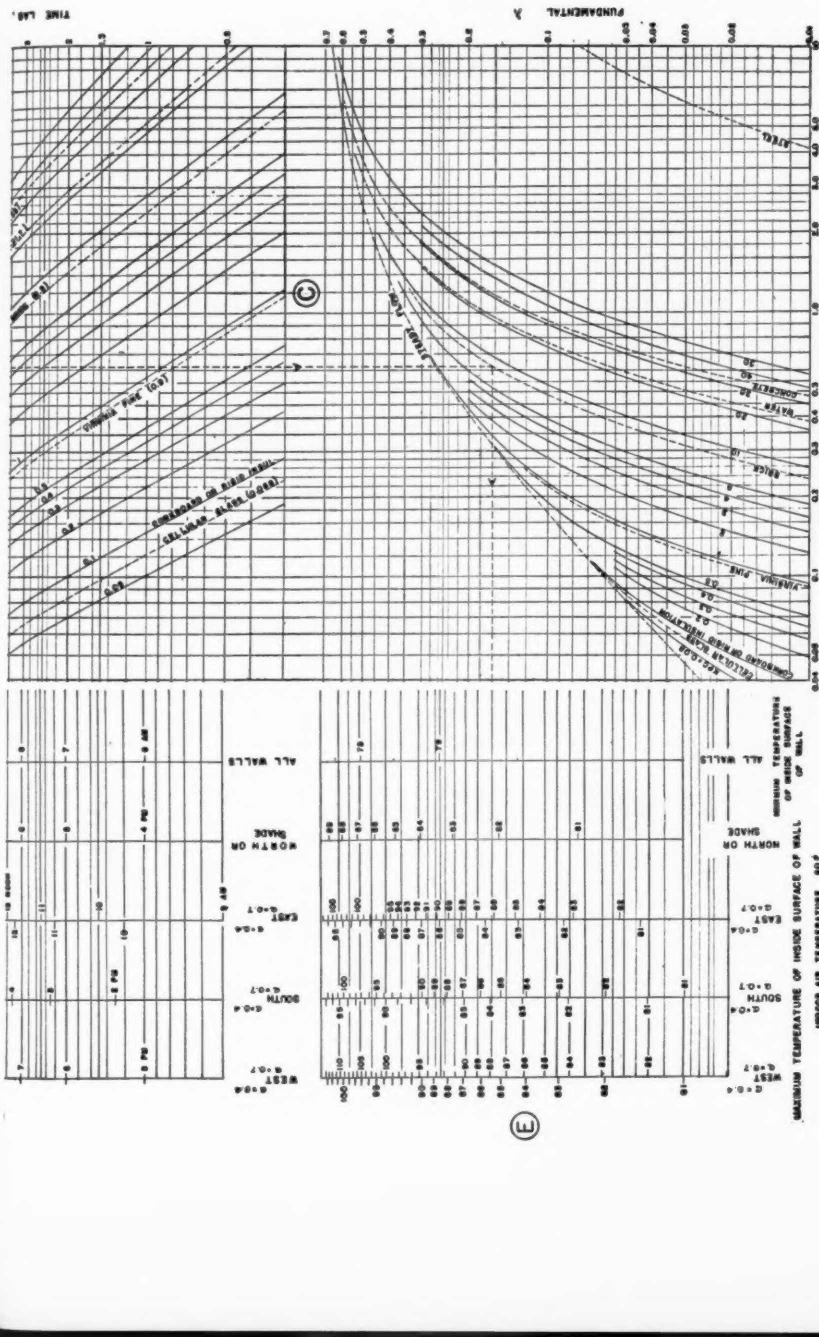


Fig. 4. Chart Showing Example with Maximum Temperature of Inside Surface of 86.3 F and a Minimum of 79.4 F

TABLE 2—APPROXIMATE SOLUTION FOR INSIDE SURFACE TEMPERATURE (16 IN. BRICK WEST WALL)

TIME OF DAY θ_i	EQUIVALENT OUTDOOR AIR TEMPERATURE (FIG. 1) t_{ei}	TIME OF DAY FOR CORRESPONDING INSIDE SURFACE TEMPERATURE $\theta_i + \frac{\phi_0}{15}$	APPROXIMATE TEMPERATURE OF INSIDE WALL SURFACE (EQ. 13) t_0	EXACT TEMPERATURE OF INSIDE WALL SURFACE; CALCULATED FROM COMPLETE SERIES SOLUTION. *See* APPENDIX A. (FIG. 2)
12 noon	93.7	12 midnight	80.4	81.7
1 P.M.	108.3	1 A.M.	80.8	81.8
2	120.5	2	81.2	81.9
3	128.1	3	81.4	81.9
4	129.4	4	81.5	81.8
5	121.5	5	81.2	81.7
6	101.2	6	80.6	81.6
7	86.7	7	80.2	81.4
8	81.0	8	80.0	81.2
9	78.1	9	79.9	81.0
10	75.2	10	79.9	80.9
11	72.5	11	79.8	80.7
12	69.9	12 noon	79.7	80.6
1 A.M.	67.7	1 P.M.	79.6	80.4
2	66.0	2	79.6	80.4
3	65.0	3	79.6	80.3
4	65.2	4	79.6	80.3
5	66.7	5	79.6	80.3
6	69.2	6	79.7	80.4
7	73.0	7	79.8	80.6
8	77.5	8	79.9	80.8
9	82.2	9	80.1	81.1
10	86.9	10	80.2	81.3
11	90.8	11	80.3	81.6

resistance ratio (λ_0) is 0.0297. Values read from the equivalent outdoor air temperature curve, Fig. 1, and approximate values for the temperature of the inside surface of the wall at the stated times calculated from Equation 13 are shown in Table 2. For comparison, exact values of the temperatures of the inside surface of the wall, as found from the complete series solutions, are given in the last column of the table.

In a case which is a particularly severe test due to the great departure from

TABLE 3—THERMAL PROPERTIES OF MATERIAL REPRESENTED IN FIG. 4 AND FIG. 5

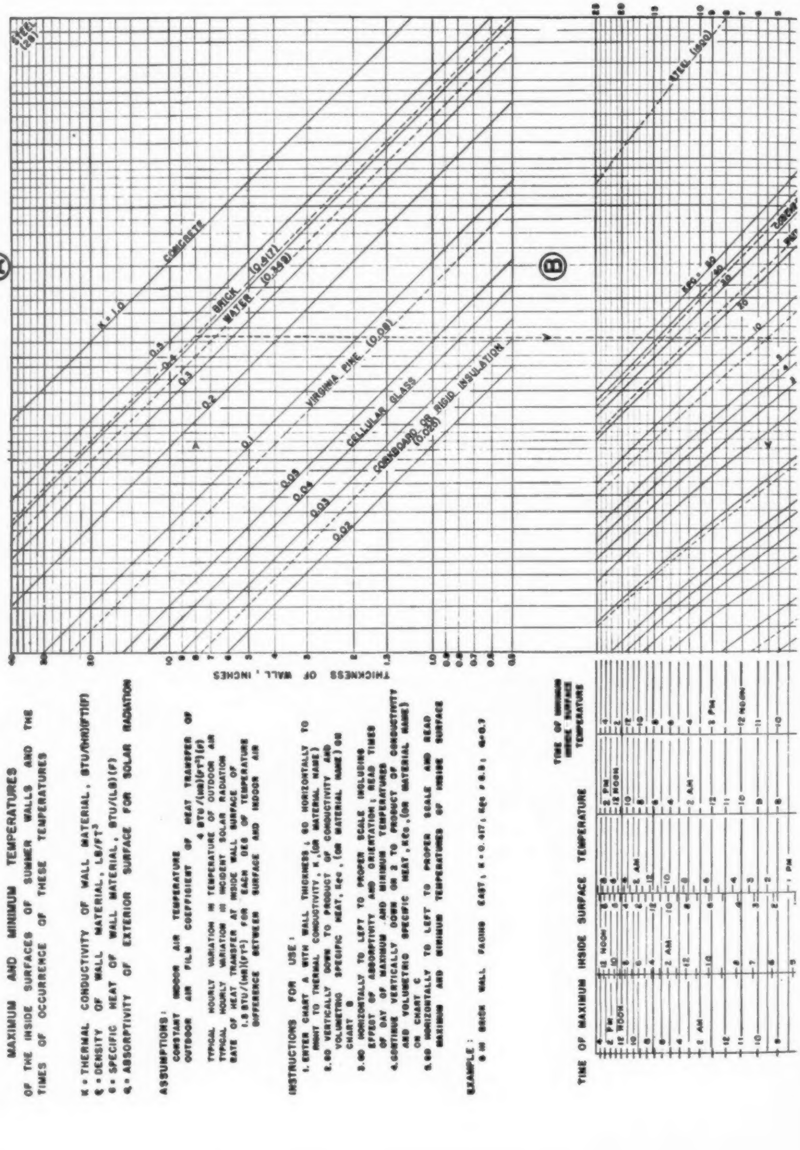
MATERIAL	THERMAL CONDUCTIVITY k BTU/(HR)(FT)(F)	VOLUMETRIC SPECIFIC HEAT ρc BTU/(FT³)(F)	PRODUCT $k\rho c$ BTU²/(HR)(FT⁴)(F²)
Cellular Glass	0.04	1.71	0.0684
Corkboard and Rigid Insulation	0.025	4.0	0.10
Virginia Pine	0.08	11.2	0.896
Brick (low-density)	0.417	19.9	8.3
Water	0.349	62.3	21.7
Concrete	1.0	35	35
Steel	28	57	1600

steady flow conditions, the approximate method is seen to give a maximum temperature of the inside wall surface that is only 0.4 F less than the true maximum and at substantially the same time of day as the true occurrence; the minimum temperature is 0.7 F less than the true minimum at substantially the time of day of the true occurrence.

To eliminate the calculations and to give a picture of the effects of the thermal properties of the wall materials on cooling load and comfort, the charts in Figs. 4 and 5 have been prepared. In these two figures, charts A, B, and C are exactly the same; these three parts are a solution of Equation 6 and Equation 5 of this paper and may be used to find the fundamental time lags and fundamental equivalent thermal resistance ratios for a single-layer wall of any material in any thickness if the thickness, thermal conductivity, and volumetric specific heats of the wall are known. These results are ***entirely independent*** of the exact variation with respect to time of outdoor air temperature and incident solar radiation. The only assumptions made in the preparation of these three parts are that the outdoor air film coefficient of heat transfer (h_L) is 4.0 Btu/(hr)(ft^2)(F) and that the rate of heat transfer at the inside surface of the wall is 1.5 Btu/(hr)(ft^2) for each degree of difference between the temperature of the inside surface of the wall and the temperature of the indoor air. Dotted lines are shown on these charts for cellular glass, corkboard or rigid insulating board, Virginia pine, low-density brick, water, concrete, and steel. The thermal properties assumed for these materials are given in Table 3. In case the user of the chart does not agree with the values assumed for these materials, he may substitute his own values for k and for $k\rho c$ and follow the solid lines.

In Fig. 4 are also given the maximum and minimum temperatures of the inside surface of the wall (chart E) and the times of occurrence (chart D) of these temperatures for a constant temperature of the indoor air of 80 F and for walls with an absorptivity for solar radiation of 0.4 and 0.7 facing East, South, and West, and for a wall completely shielded from solar and sky radiation. The value of solar absorptivity of 0.4 corresponds to extremely light colors of the exterior surface of the wall and 0.7 to medium dark colors.

In Figs. 4 and 5, the extremes of surface temperature (chart E) and the times of occurrence of these extremes (chart D) are based upon the variation of equivalent outdoor air temperatures shown in Fig 6. This figure gives different results than those shown in Fig. 1. The values of Fig. 1 were based upon Weather Bureau observations for Ithaca, N. Y., where a range in the outdoor dry-bulb temperature of 30 F is common on a summer day when the maximum temperature is equal to that commonly used in selecting cooling equipment. From a study of Summer Weather Data published by the Marley Co. it was seen that the daily range of dry-bulb temperature on a design day in many cities is under 20 F; for example, the daily range of dry-bulb temperature in Pittsburgh, where tests on wall panels were made by the A.S.H.V.E. Research Laboratory, is usually less than 30 F. The dry-bulb temperature of the outdoor air assumed in Fig. 6 is the one given for New York in Summer Weather Data; this curve gives a maximum outdoor air temperature of 93.5 F at 3 p.m. and a minimum outdoor air temperature of 76 F at 5 a.m. The incident solar radiation used in Fig. 6 includes direct solar radiation and sky radiation and is that given in the A.S.H.V.E. Guide 1942 as typical for August 1


MAXIMUM AND MINIMUM TEMPERATURES
OF THE INSIDE SURFACES OF SUMMER WALLS AND THE TIMES OF OCCURRENCE OF THESE TEMPERATURES
K = THERMAL CONDUCTIVITY OF WALL MATERIAL, BTU/(HR)(FT)(F)
ρ = DENSITY OF WALL MATERIAL, LB/FT³
c = SPECIFIC HEAT OF WALL MATERIAL, BTU/(LB)(F)
a = ABSORPTIVITY OF EXTERIOR SURFACE FOR SOLAR RADIATION
ASSUMPTIONS:
CONSTANT INDOOR AIR TEMPERATURE
OUTDOOR AIR FILM COEFFICIENT OF HEAT TRANSFER OF 4 BTU/(HR)(FT²)(F)
TYPICAL HOURLY VARIATION IN TEMPERATURE OF OUTDOOR AIR
TYPICAL HOURLY VARIATION IN INCIDENT SOLAR RADIATION
RATE OF HEAT TRANSFER AT INSIDE WALL SURFACE OF 1.8 BTU/(HR)(FT²) FOR EACH DEG OF TEMPERATURE DIFFERENCE BETWEEN SURFACE AND INDOOR AIR
INSTRUCTIONS FOR USE:
1. ENTER CHART A WITH WALL THICKNESS; GO HORIZONTALLY TO RIGHT TO THERMAL CONDUCTIVITY, K, (OR MATERIAL NAME)
2. GO VERTICALLY DOWN TO PRODUCT OF CONDUCTIVITY AND VOLUMETRIC SPECIFIC HEAT, Kρc, (OR MATERIAL NAME) ON CHART B
3. GO HORIZONTALLY TO LEFT TO PROPER SCALE INCLUDING EFFECT OF ABSORPTIVITY AND ORIENTATION; READ TIMES OF DAY OF MAXIMUM AND MINIMUM TEMPERATURES
4. CONTINUE VERTICALLY DOWN ON 2 TO PRODUCT OF CONDUCTIVITY AND VOLUMETRIC SPECIFIC HEAT, Kρc, (OR MATERIAL NAME) ON CHART C
5. GO HORIZONTALLY TO LEFT TO PROPER SCALE AND READ MAXIMUM AND MINIMUM TEMPERATURES OF INSIDE SURFACE
EXAMPLE:
8 IN BRICK WALL FACING EAST; K = 0.417; Kρc = 8.8; a = 0.7
A
STEEL (26)
K = 1.0
CONCRETE
0.5
0.4
0.3
BRICK (0.417)
WATER (0.348)
0.2
0.1
VIRGINIA PINE (0.08)
0.05
0.04
0.03
0.02
CELLULAR GLASS
CORKBOARD OR RIGID INSULATION (0.025)
THICKNESS OF WALL, INCHES
B
STEEL (1900)
TIME OF MAXIMUM INSIDE SURFACE TEMPERATURE
TIME OF MINIMUM INSIDE SURFACE TEMPERATURE
D

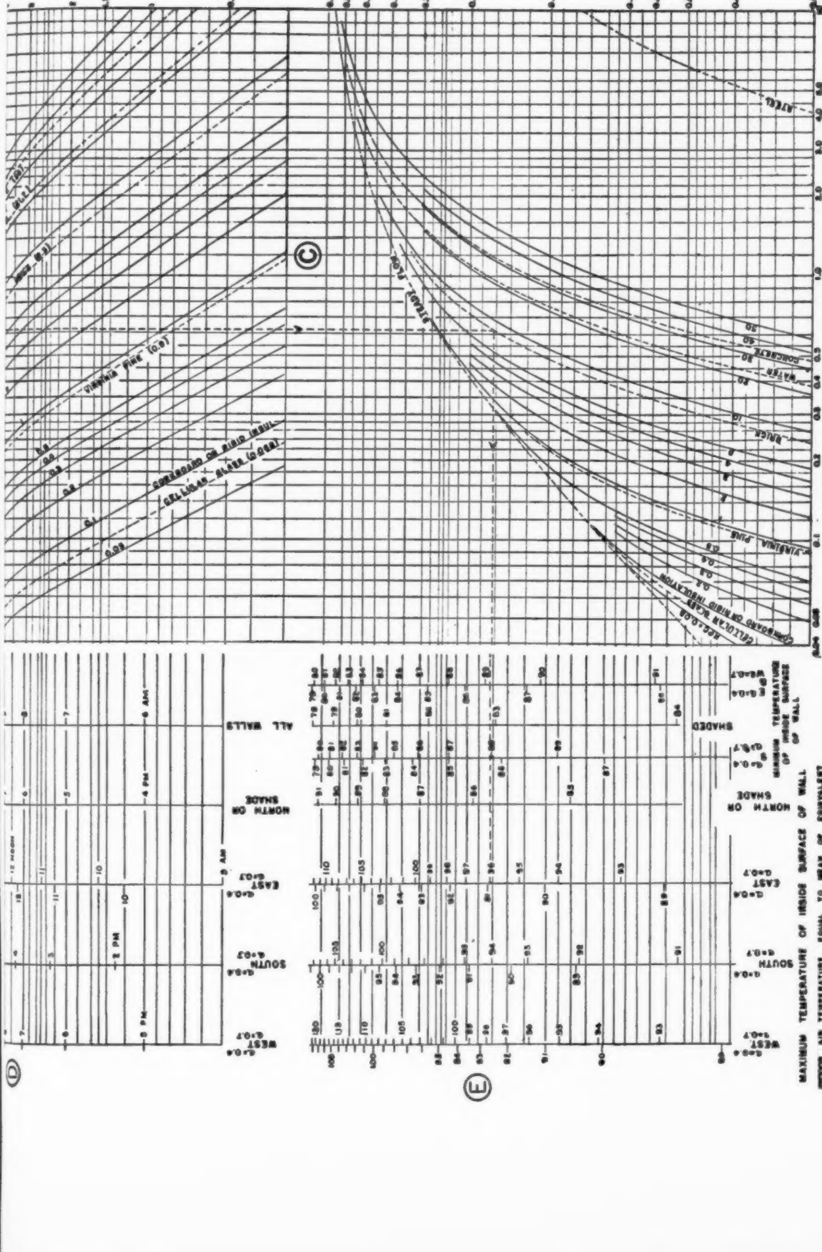


Fig. 5. Chart Showing Example with Maximum Temperature of Inside Surface of 96.0 F and a Minimum of 89.1 F

and a North latitude of 40 deg. Values of equivalent outdoor air temperatures based upon these data for absorptivities of 0.4 and 0.7 are given in Table 4.

Another use of Fig. 4 is in the correlation of the effects of thickness and properties of the wall materials on cooling load. In his analysis of the cooling load, the air-conditioning engineer is interested in the rate of heat transfer from the inside surface of the walls at other times than those of maximum and minimum surface temperatures, and this additional information is also needed in a complete study of comfort. Such information may be obtained by using the equivalent outdoor air temperature (Fig. 6), charts A, B, and C, and the approximate method of solution for surface temperature. From Equation 13, the approximate temperature of the inside surface of the wall at any time θ_1 is

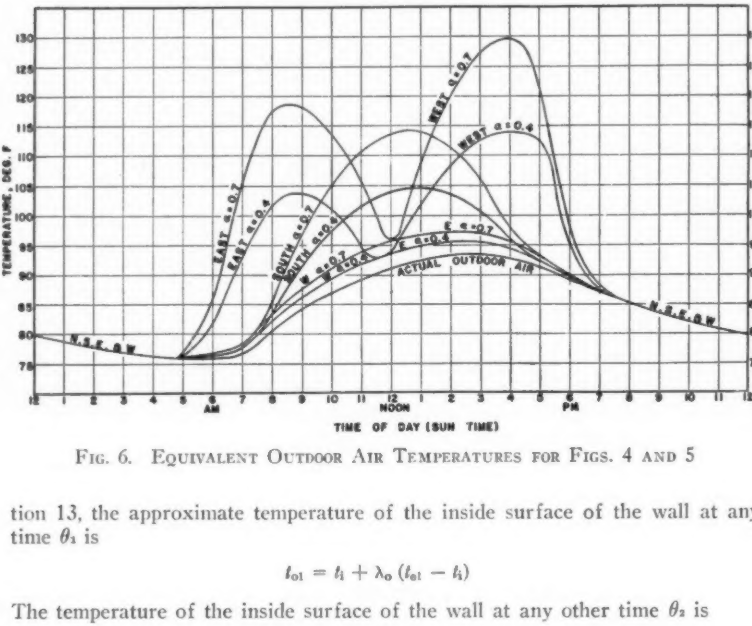


FIG. 6. EQUIVALENT OUTDOOR AIR TEMPERATURES FOR FIGS. 4 AND 5

$$t_{o1} = t_i + \lambda_o (t_{e1} - t_i)$$

The temperature of the inside surface of the wall at any other time θ_2 is

$$t_{o2} = t_i + \lambda_o (t_{e2} - t_i)$$

Therefore,

$$t_{o1} - t_{o2} = \lambda_o (t_{e1} - t_{e2})$$

To find the temperature of the inside surface of the wall at any time, subtract from the maximum temperature of the inside surface the product of λ_o (chart C) and the difference between the *maximum* equivalent outdoor air temperature and the equivalent outdoor air temperature at that time, after making a correction for the time lag (chart B). The corresponding rate of heat transfer from the inside surface of the wall in Btu/(hr)(ft²) is

$$q = h_o^r (t_o - t_i) = 1.5 (t_o - t_i)$$

This procedure will next be shown by an illustrative example. It is desired to find the hourly variation in the temperature of the inside surface and the rate of heat flow from that surface of an 8-in. brick (common low-density) wall facing South for a constant indoor air temperature of 80 F. The properties used for brick are those given in Table 3, the solar absorptivity will be taken as 0.7. From Fig. 4, with $\frac{k}{L} = 0.625$, the time lag is 5.5 hours and $\lambda_o = 0.163$. From Fig. 4, the maximum temperature of the inside surface of the wall is 85.6 F occurring about 6 hours after noon. From Fig. 6, the maximum equivalent outdoor air temperature is 114.2 R occurring about 0.5 hours after noon. At 7 hours after noon, then, the temperature of the inside surface of the wall will be 85.6 F minus the product of λ_o and the difference between the equivalent outdoor air temperatures at 0.5 hour and at 1.5 hours after noon, or

TABLE 4—EQUIVALENT OUTDOOR AIR TEMPERATURES (*see* FIG. 6). ACTUAL OUTDOOR AIR TEMPERATURE FROM SUMMER WEATHER DATA (NEW YORK). INCIDENT SOLAR AND SKY RADIATION FROM A.S.H.V.E. GUIDE 1932. (40 DEG N. LAT., AUG. 1.) ASSUMED OUTDOOR AIR FILM COEFFICIENT OF HEAT TRANSFER = 4 BTU/(HR)(FT²)(F)

SUN TIME	OUTDOOR DRY-BULB TEMPERATURE F	EQUIVALENT OUTDOOR TEMPERATURE, F					
		Absorptivity = 0.4 Wall Facing			Absorptivity = 0.7 Wall Facing		
		E	S	W	E	S	W
12 noon	91.2	93.8	104.0	93.8	95.8	113.6	95.8
1 P.M.	92.4	94.9	104.8	101.8	96.9	114.1	108.8
2 P.M.	93.2	95.5	103.5	108.4	97.3	111.2	119.8
3 P.M.	93.5	95.6	100.9	113.0	97.1	106.4	127.6
4 P.M.	92.9	94.6	95.8	114.0	95.9	97.8	129.8
5 P.M.	91.4	92.5	92.5	107.6	93.1	93.1	120.8
6 P.M.	89.0	89.4	89.4	94.6	89.8	89.8	98.8
7 P.M.	87.0	87.2	87.2	87.6	87.4	87.4	88.1
8 P.M.	85.3	85.3	85.3	85.3	85.3	85.3	85.3
9 P.M.	83.6	83.6	83.6	83.6	83.6	83.6	83.6
10 P.M.	82.1	82.1	82.1	82.1	82.1	82.1	82.1
11 P.M.	80.9	80.9	80.9	80.9	80.9	80.9	80.9
12 midnight	79.8	79.8	79.8	79.8	79.8	79.8	79.8
1 A.M.	78.9	78.9	78.9	78.9	78.9	78.9	78.9
2 A.M.	77.9	77.9	77.9	77.9	77.9	77.9	77.9
3 A.M.	77.1	77.1	77.1	77.1	77.1	77.1	77.1
4 A.M.	76.4	76.4	76.4	76.4	76.4	76.4	76.4
5 A.M.	76.0	76.6	76.2	76.2	76.1	76.4	76.4
6 A.M.	76.0	81.6	76.4	76.4	85.8	76.8	76.8
7 A.M.	76.6	92.8	77.7	77.7	106.0	78.3	78.3
8 A.M.	80.6	101.7	83.5	72.3	117.5	85.5	83.6
9 A.M.	84.2	103.7	91.6	86.3	118.3	97.1	87.6
10 A.M.	87.0	102.2	97.3	89.3	113.6	105.0	91.1
11 A.M.	89.3	94.7	101.7	91.8	105.7	111.0	93.8
12 noon	91.2	93.8	104.0	93.8	95.8	113.6	95.8

$$t_o = 85.6 - 0.163\ (114.2 - 112.8) = 85.4\ \text{F}.$$

The remainder of the calculation is shown in Table 5 and the results are plotted in Fig. 7. Obviously, results may be found for other walls of any thickness, thermal properties, and orientation in a similar manner. It should be emphasized that the times used throughout this paper are sun times. Although the solution is approximate, a comparison with exact results, as previously shown in this paper, will show that the error is not greater than that usually introduced due to lack of *precise* knowledge concerning the important thermal properties of the wall materials.

TABLE 5—TEMPERATURES OF INSIDE SURFACE AND HEAT TRANSFER RATES FOR 8-IN. SUNLIT BRICK WALL FACING SOUTH

TIME OF DAY HOURS AFTER NOON θ_1	EQUIVALENT OUTDOOR AIR TEMPERATURE (FIG. 4)	TIME OF DAY OF CORRESPONDING INSIDE SURFACE TEMPERATURE HOURS AFTER NOON $\theta_2 = \theta_1 + $ (LAG)	TEMPERATURE OF INSIDE SURFACE OF WALL AT θ_2	RATE OF HEAT TRANSFER FROM INSIDE SURFACE AT θ_2 BTU/(HR)(FT²)
0.5	114.2	6	85.6	8.40
1.5	112.8	7	85.4	8.10
2.5	109.1	8	84.8	7.20
3.5	101.7	9	83.6	5.40
4.5	95.4	10	82.5	3.75
5.5	91.6	11	81.9	2.85
6.5	88.6	12	81.4	2.10
7.5	86.2	13	81.0	1.50
8.5	84.4	14	80.7	1.05
9.5	82.7	15	80.5	0.75
10.5	81.4	16	80.3	0.45
11.5	80.2	17	80.1	0.15
12.5	79.2	18	79.9	−0.15
13.5	78.4	19	79.8	−0.30
14.5	77.4	20	79.6	−0.60
15.5	76.6	21	79.5	−0.75
16.5	76.1	22	79.4	−0.90
17.5	76.5	23	79.5	−0.75
18.5	77.5	24	79.8	−0.30
19.5	81.0	25	80.2	0.30
20.5	91.5	26	81.9	2.85
21.5	101.2	27	83.5	5.25
22.5	108.1	28	84.7	7.05
23.5	112.5	29	85.3	7.95

The extremes of surface temperature (chart E) and the times of their occurrence (chart D) of Fig. 5 are for the case where no cooling equipment is operated within the room and are based upon a constant indoor air temperature equal to the mean equivalent outdoor air temperature for each orientation; in this case the net daily gain of heat through the wall is zero. The *only difference* between Figs. 4 and 5, then, is that Fig. 4 is drawn for a constant indoor air temperature of 80 F, while Fig. 5 is for the constant indoor air temperature given in Table 6.

The effect of thickness and thermal properties of the wall upon the extreme temperatures and upon time lag, shown in Fig. 5, may be used in the selection of walls for uncooled enclosures with the purpose of providing the maximum

comfort during the time of occupancy. For example, assume that a single layer wall is to be selected which will give a minimum temperature of the inside surface near the middle of the day, say, at 12 noon. Regardless of orientation, chart D of Fig. 5 shows that the time lag must be 7 hours. Following this time lag into chart B and up into chart A, the thickness of the wall of

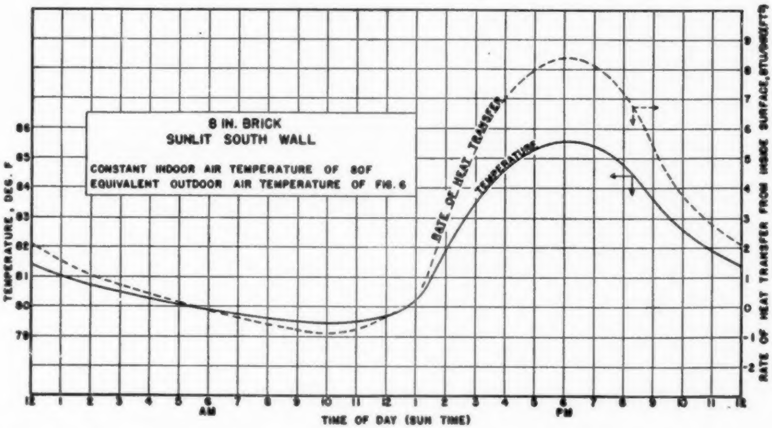


FIG. 7. TEMPERATURE OF INSIDE SURFACE AND RATE OF HEAT TRANSFER TO INDOOR AIR FOR 8-IN. BRICK SOUTH WALL

any specified material may be found; from the fundamental λ of chart C, the actual minimum temperature of each inside surface may be read on the scale of chart E or calculated. Results of the comparison are shown in Table 7.

SUMMARY OF RESULTS

1. Charts are presented that give the maximum and minimum temperatures, and the times of occurrence of these temperatures, for single layer walls of any material, thickness, and principal orientation, when the temperature of the indoor air is constant.

2. A simple, approximate method of finding the effects of thickness and thermal properties of wall materials upon the hourly variation in the rate of heat flow at the inside surface is given for single-layer walls with any hourly variation in the outdoor air temperature and in the incident solar radiation and for any assumed constant temperature of the indoor air. This method is as accurate as the usual knowledge of the thermal properties of the wall materials and may be used to correlate experimental results previously obtained. Actual experimental tests on this problem are now being conducted at the John B. Pierce Laboratory of Hygiene.

3. The effects of thermal properties and thickness of wall materials upon conditions influencing comfort in an uncooled enclosure are discussed and examples given. The choice of the thermal properties and thickness of the wall materials for comfort is shown to be intimately associated with the orientation and time of occupancy of a room. It is shown that a building with all walls of the same material in a uniform thickness does not meet the ideal thermal requirements.

TABLE 6—CONSTANT INDOOR AIR TEMPERATURE FOR ZERO HEAT TRANSFER, DAILY (FOR THE EQUIVALENT OUTDOOR AIR TEMPERATURES OF FIG. 6 AND TABLE 4)

ORIENTATION	CONSTANT INDOOR AIR TEMPERATURE SOLAR ABSORPTIVITY	
	0.4	0.7
Completely Shielded from Solar and Sky Radiation	84.3 F	84.3 F
South	87.7	90.3
East and West	88.4	91.6

LIMITATIONS OF RESULTS AND SUGGESTIONS FOR FUTURE STUDY

All results given in this paper are for walls of a single material. A similar study of composite walls would be useful. In this connection, it should be pointed out that the results for a composite wall cannot be found accurately from the results for a single-layer wall, because the order of sequence in which the temperature wave encounters the layers of a composite wall will influence the time lag and the flow of heat.[3]

Charts A, B, and C of Figs. 4 and 5 are completely general except for the assumptions made for the value of the outdoor air film coefficient of heat transfer of 4.0 Btu/(hr) (ft^2) (F) and for the rate of heat transfer at the inside surface of 1.5 Btu/(hr) (ft^2) for each degree of difference between the temperature of the inside surface and that of the indoor air. There can be little criticism of using the value assumed for the outdoor air film coefficient of heat transfer; a value of 4 is better for the usual summer wind velocities than the value of 6 which is commonly assumed for winter wind velocities. At the same time, the exterior surface of a building wall may lose heat by radiating to the cold sky on a clear night. It would be difficult to correct accurately for this effect, and the only practical possibility is in lowering the assumed equivalent outdoor air temperature during the evening; for this reason, some may prefer the equivalent outdoor air temperature assumed in Fig. 1 to that of Fig. 6.

The value assumed for the rate of heat transfer at the inside wall surface is based upon a transfer of heat by convection to the room air at the rate of 0.5 Btu/(hr)

TABLE 7—COMPARISON OF WALL MATERIALS THAT GIVE A MINIMUM TEMPERATURE OF THE INSIDE WALL SURFACE AT 12 NOON (FIG. 5)

MATERIAL	CELLULAR GLASS	CORKBOARD OR RIGID INSULATION	VIRGINIA PINE	BRICK (LOW DENSITY)	CONCRETE
$\frac{k}{L}$ (Btu/hr ft^2 F)	0.037	0.045	0.142	0.504	1.23
k (Btu/hr ft F)	0.04	0.025	0.08	0.417	1.0
Thickness (inches)	13	6.7	6.8	9.9	9.8
λ_0	0.01043	0.0155	0.043	0.108	0.175
Min. Temp., F, West Wall, a = 0.7	91.4	91.3	90.9	89.9	88.9
Max. Temp., F, West Wall, a = 0.7	92.0	92.2	93.2	95.7	98.3
Time of Max. Temp., West Wall, a = 0.7	11 P.M.	11 P.M.	11 P.M.	11 P.M.	11 P.M.

[3] Transient Heat Load in Calculating Air Conditioning Loads, by E. M. Pugh. (Refrigerating Engineering, July, 1941.)

(ft²) (F), a transfer of heat by radiation to solid surfaces seen by the wall at the rate of 1.0 Btu/(hr) (ft²) (F), and a temperature of all solid surface *seen* by the wall equal to that of the indoor air. The film coefficient of heat transfer by convection from large vertical surfaces to *still* air is commonly taken either as $0.27(\Delta t)^{1/4}$ or $0.22(\Delta t)^{1/3}$. For small temperature differences, the rate of heat exchange by radiation between solid surfaces may be taken proportional to the temperature difference instead of to the difference between the fourth powers of the absolute temperatures; for surfaces like the usual materials of construction, which have absorptivities for low-temperature radiation between 0.9 and 1.0, the rate of heat exchange by radiation between two surfaces may be calculated from the following equation:

$$q = A_1 F_{12} (t_1 - t_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (14)$$

where

q = rate of heat exchange, Btu/hr
A_1 = area of surface 1, ft²
F_{12} = angle factor, fraction of radiant energy originating at surface 1 which is intercepted at 2
t_1 = temperature of surface 1, F
t_2 = temperature of surface 2, F

When all the surfaces in the room, except the one wall panel, are at the temperature of the room air and when these surfaces have the usual high absorptivities, the rate of heat transfer at the inside surface of the wall is:

$$\frac{q}{A(t_o - t_i)} = 0.22(t_o - t_i)^{1/3} + 1$$

For temperature differences between inside surface and indoor air of 10 to 15 F, the right-hand side of this equation reduces to about 1.5 Btu/(hr) (ft²) (F).

Possible exceptions to this rule may be noted. If the inside surfaces of the wall are completely reflecting, the rate of heat transfer at the inside surface will be about 0.5 Btu/(hr) (ft²) (F). Depending upon the wall material, and with all other conditions the same, this lower rate of heat transfer at the inside surface may increase the daily range in the temperature of that surface by from 50 to 150 per cent when compared with the results of this paper; lowering the rate of heat transfer at the inside surface depresses the daily minimum and raises the daily maximum temperature of that surface. Another exception occurs with absorbing surfaces when the temperatures of all surfaces in the room, except the wall under consideration, are not equal to that of the indoor air. For example, consider a cooled room with two perpendicular interior walls and two interior partitions; for simplicity, assume there are no windows. Let the instantaneous temperature of the indoor air be 80 F, that of the inside surface of the wall under consideration ($A_1 = 96$) be $t_1 = 85$ F, that of the inside surface of the other exterior wall ($A_2 = 120$) be $t_2 = 90$ F, and that of all other interior surfaces ($A_3 = 576$) be $t_3 = 80$ F. Reasonable angle factors for this case are $F_{12} = 0.18$ and $F_{13} = 0.82$. The rate of heat loss by convection from surface 1 to room air is

$$\frac{q}{A_1} = 0.22(5)^{1/3} = 0.38 \text{ Btu/(hr) (ft}^2\text{)}$$

The rate of heat loss by radiation from surface 1 is

$$\frac{q}{A_1} = 0.82(5) - 0.18(5) = 3.2 \text{ Btu/(hr) (ft}^2\text{)}$$

The rate of heat loss by convection and radiation from surface 1 per degree of temperature difference between that surface and the room air is

$$\frac{q}{A_1(t_o - t_i)} = \frac{3.58}{5} = 0.72 \text{ Btu/(hr) (ft}^2\text{)}$$

Generalizing from this example, it is obvious that no simple mathematical procedure, no simple electrical analogy test, and no wall panel in a test cubicle will reproduce *exactly* the conditions that may exist in *all* rooms. The results presented in this paper and in all other papers on the same subject must be interpreted and used with these shortcomings clearly in mind.

In Figs. 4 and 5, the thermal properties assigned to the materials named on the charts are given in Table 3. These properties must not be considered as absolute and invariable. A brief study of the literature that might be consulted by an engineer interested in the estimate of cooling loads shows the tremendous range in the values of important thermal properties that he might encounter. For example, study of the International Critical Tables, HEATING, VENTILATING, AIR CONDITIONING GUIDE, and *ASRE Data Book,* only, yields the following range in the values of the thermal properties of three common materials of construction at room temperatures:

MATERIAL	APPARENT DENSITY LB/FT^3	THERMAL CONDUCTIVITY BTU/(HR)(FT)(F)	SPECIFIC HEAT BTU/(LB)(F)	POSSIBLE RANGE IN $k\rho c$ (BTU)^2/(HR)(FT^4)(F^2)
Brick (Masonry)	90—140	0.03—0.8	0.20—0.22	5.4 —24.6
Concrete........	135—150	0.45—1.36	0.16—0.27	9.7 —55.1
Pine...........	22—40	0.05—0.09	0.32—0.65	0.35— 2.34

Data for specific heats of building materials are the most incomplete. A collection in one table of accepted values of the apparent density, thermal conductivity, and specific heat of common materials of construction would prove useful to the engineer interested in problems in unsteady flow of heat. Also, more information is necessary concerning the absorptivity for solar energy of the outside wall surfaces.

APPENDIX A

Details of Calculations for Results Shown in Fig. 2 and Fig. 3. Fourier series for outdoor air temperature in Fig. 1:

$$t_a = 80 + 14.86\cos(15\theta - 37) + 1.79\cos(30\theta - 353) + 0.734\cos(45\theta - 159) + 0.458\cos(60\theta - 240) \quad . . . (15)$$

Fourier series for incident solar radiation in Fig. 1:

East Wall
$$I = 37.6 + 69.8\cos(15\theta - 306) + 55\cos(30\theta - 252) + 35.2\cos(45\theta - 197) + 16.5\cos(60\theta - 139) \quad . . . (16)$$

South Wall
$$I = 23.1 + 41\cos(15\theta) + 28\cos(30\theta) + 13\cos(45\theta) \quad . . . (17)$$

West Wall
$$I = 37.6 + 69.8\cos(15\theta - 54) + 55\cos(30\theta - 108) + 35.2\cos(45\theta - 163) + 16.5\cos(60\theta - 221) \quad . . . (18)$$

Examples of the equations for the temperatures of the inside surface of the walls shown in Fig. 2:

West Sunlit Wall
16 = in. Brick:

$$t_o = 81.07 + 0.442\cos(15\theta - 217) + 0.014\cos(30\theta - 259) + 0.363\cos(15\theta - 234) + 0.077\cos(30\theta - 14) \quad . . . (19)$$

1½ = in. Cellular Glass:

$$t_o = 81.09 + 2.45\cos(15\theta - 40) + 0.295\cos(30\theta - 358) + 2.014\cos(15\theta - 57) + 1.585\cos(30\theta - 113) + 1.013\cos(45\theta - 171) \quad . . . (20)$$

	Wall Thickness and Material	
	16-in. Brick	1½-in. Cellular Glass
Equivalent thermal resistance ratio, λ		
Fundamental, λ_0	0.0297	0.1649
First harmonic, λ_1	0.00796	0.1645
Second harmonic, λ_2	0.00293	0.1640
Third harmonic, λ_3	....	0.1636
Lag angle, ϕ		
Fundamental, ϕ_0	180 deg	2.6 deg
First harmonic, ϕ_1	266 "	5.1 "
Second harmonic, ϕ_2	330 "	7.7 "
Third harmonic, ϕ_3	....	10.3 "

Examples of the equations for the temperatures of the inside surfaces of the walls shown in Fig. 3:

West Sunlit Wall

16-in. Brick: add 5:51 F to results obtained from Equation 19.

1½-in. Cellular Glass: add 5.49 F to results obtained from Equation 20.

Acknowledgment

The authors wish to acknowledge the support and interest of the John B. Pierce Foundation which made possible the study presented in this paper. In particular, they acknowledge the helpful ideas and interest of R. L. Davison, Director of Research, John B. Pierce Foundation. Valuable assistance in the calculations and preparation of the charts was given by R. E. Clark, Assistant Professor of Heat-Power Engineering, and N. R. Gay, Instructor of Heat-Power Engineering, both of Cornell University.

DISCUSSION

L. T. Wright, Jr. (Written): In the complete mathematical solution it is evident that, for fixed values of h_o and h_L, the decrement factor, λ_n, and the lag angle, ϕ_n, are functions only of $k\sigma_n$ and $\sigma_n L$.

Assume a material with properties designated by the subscript 1 for which it is desired to find λ_{1n} and ϕ_{1n} for harmonics higher than the first. For this material

$$\sigma_{1n}L_1 = \sqrt{\frac{0.1309n(\rho c)_1 L_1^2}{k_1}} \qquad (1)$$

and

$$k_1\sigma_{1n} = \sqrt{0.1309nk_1(\rho c)_1} \qquad (2)$$

For each higher harmonic we wish to find an equivalent material which has such thermal properties and thickness that its *fundamental* decrement factor and lag angle are equal respectively to the *higher order* decrement factor and lag angle of material 1. Designate the properties and thickness of this equivalent material by the subscript *e*. Now,

$$\sigma_{e1}L_e = \sqrt{\frac{0.1309(\rho c)_e L_e^2}{k_e}} \quad \text{. . . . .} \quad (3)$$

and

$$k_e\sigma_{e1} = \sqrt{0.1309k_e(\rho c)_e} \quad \text{. . . . .} \quad (4)$$

By equating Equations (2) and (4) we obtain the following:

$$k_e(\rho c)_e = nk_1(\rho c)_1 \quad \text{. . . . .} \quad (5)$$

By equating Equations (1) and (3) we obtain the following:

$$\frac{(\rho c)_e L_e^2}{k_e} = n\frac{(\rho c)_1 L_1^2}{k_1} \quad \text{. . . . .} \quad (6)$$

Divide Equation (6) by Equation (5) to obtain

$$\left(\frac{L_e}{k_e}\right)^2 = \left(\frac{L_1}{k_1}\right)^2 \quad \text{. . . . .} \quad (7)$$

Now examine Equations (5) and (7). Choose $k_e = k_1$; $L_e = L_1$; and $(\rho c)_e = n(\rho c)_1$. Then $\lambda_{e1} = \lambda_{1n}$ and $\phi_{e1} = \phi_{1n}$.

It should be pointed out that a much more satisfactory approximate equation than Equation (13) for finding the inside surface temperature at any time of the day can be given. This equation is

$$t_{o\left(\theta+\frac{\phi_o}{15}\right)} = t_i + \frac{\frac{1}{h_0}\left(t_{em} - t_i\right)}{\frac{1}{h_0} + \frac{L}{k} + \frac{1}{h_i}} + \lambda_0\left(t_{e\theta} - t_{em}\right)$$

Equation (13) of the paper was given only because it gave a simple method of plotting Charts D and E of Figs. 4 and 5. These charts are reasonably accurate, but not as accurate as the results calculated from the equation just given.

VICTOR PASCHKIS, New York, N. Y. (WRITTEN): This paper contains very interesting charts for the calculation of the periodic flow of heat through single layer walls. The charts are, however, as I understand, based on some assumptions, one of which leads to so considerable an inaccuracy as to render doubtful the practical usefulness of the charts as presented in the paper:

> The assumption that it is sufficient to work with the *fundamental* sine wave and that it is permissible to neglect all higher harmonics.

To bear out this statement, reference is made to Table 2 of the paper. The fourth column, entitled, Approximate Temperature of Inside Wall Surface (Equation 13), shows the temperatures as they could be read from the graphs, Figs. 4 and 5. These graphs are calculated on the basis of the above mentioned assumption (i.e., using the fundamental only). The fifth column, entitled, Exact Temperature of Inside Wall Surface; calculated from complete series solution (Fig. 2), is based on a calculation where this simplifying assumption is not made.

At first sight, the two columns appear to check very closely. However, the point of interest is of course not the inside wall surface temperature, but the heat flow from the inside surface to the (constant) inside air temperature. The inside film

conductance is 1.5 Btu/(ft^2) (hr) (F). The heat flow is then 1.5 times the difference between the inside wall surface temperature and the constant inside air temperature. This air temperature is assumed to be 80 F.

The accuracy of the charts can thus be determined as follows: deduct from every value in columns 4 and 5 the constant inside air temperature of 80 F; compare the differences.

Thus the following figures and facts were found:

1. The approximate values lie consistently below the accurate values.
2. Only at one time (4 P.M.) is the error 20 per cent (Temperature difference 1.5 instead of 1.8). At all other times the error is much larger. That is, at 12 noon it is 425 per cent (Temperature difference 0.4 instead of 1.7).
3. The total heat flow (over the 24-hr period) following the charts (column 4) is 6.9 Btu per square foot, while the correct value (column 5) is 38.5 Btu per square foot or 5.6 times as high.

TABLE A—APPROXIMATE AND ACCURATE VALUES OF HEAT FLOW FOR VARIOUS HOURS OF DAY

TIME	VALUES OF HEAT FLOW IN BTU/SQ FT, HR FROM INSIDE WALL SURFACE TO AIR. FILM CONDUCTANCE 1.5 BTU/SQ FT, HR, F; AIR TEMPERATURE 80 F	
	Approx. Value (Equ. 13)—Charts 4 and 5	Acc. Value (Considering Higher Harmonics)
12 noon	+0.6	2.55
1 P.M.	+1.2	2.70
2 P.M.	+1.8	2.85
3 P.M.	+2.1	2.85
4 P.M.	+2.25	2.70
5 P.M.	+1.8	2.55
6 P.M.	+0.9	2.40
7 P.M.	+0.3	2.10
8 P.M.	+0	1.8
9 P.M.	−0.15	1.5
10 P.M.	−0.15	1.35
11 P.M.	−0.30	1.05
12 P.M.	−0.45	0.90
1 A.M.	−0.60	0.60
2 A.M.	−0.60	0.60
3 A.M.	−0.60	0.45
4 A.M.	−0.60	0.45
5 A.M.	−0.60	0.45
6 A.M.	−0.45	0.60
7 A.M.	−0.30	0.90
8 A.M.	−0.15	1.2
9 A.M.	+0.15	1.65
10 A.M.	+0.30	1.95
11 A.M.	+0.45	2.4
Average	+0.287	1.606

In Table A the approximate and accurate values of heat flow are tabulated for the various hours of the day. The constant inside film conductance is 1.5 Btu/(ft^2) (hr) (F); the constant inside air temperature 80 F.

It is obvious that the approximate solution yields answers which are not accurate enough for practical purposes.

The authors, incidentally, do not tell how many higher harmonics they have considered in determining the accurate solution. It would be interesting to know this; possibly the error is still larger than indicated in this discussion.

C. M. ASHLEY, Syracuse, N. Y.: This paper is of great significance for the Society. It introduces what in substance is a new technique for the solution of one of our most important problems. We should fully appreciate the fundamental significance of establishing our heat transfer values on a more sound basis.

The idea of combining the sun effect with the outside air temperature to obtain an over-all picture as to the amount of heat conveyed to the outside of the wall, is a very significant contribution. It points the way to a means by which simplified expressions for practical use in the field can be derived by using that same approach to the problem to obtain, for example a fictitious outside temperature expressing the sun load.

The separation of the wall factors from the outside factors represents a contribution. The fact that those factors, after analysis, can be expressed independently of the outside conditions, makes possible a definite analysis of all the factors which are involved in the problem. This was previously thought to be entirely beyond the scope of possibility and the best we could hope to do was to fix most of the variables and get a partial solution to the general problem.

Some of the approximations that are made in the paper are a little unfortunate. Since the paper has been published, however, the authors have found a means of improvement which does not involve much extra work. This additional work which they have done is in itself another step forward beyond the written paper. Out of this and out of a plan for a future program, which it is hoped will be put into effect during the next two or three years, we can line up the whole problem of the transmission of heat through walls and rooms in a most satisfactory manner, and include all the variables which, in effect, actually govern the process.

J. N. HADJISKY, Birmingham, Mich.: How serious a problem of that kind may become was brought to my attention a year ago, when a large building had to be designed for industrial purposes. The preliminary estimates called for 3,000 tons of refrigeration. Two or three of us engineers were delegated to make an independent study, and our estimates differed by a very large amount, chiefly because unreliable information of volumetric capacities and other constants varied a very great deal. Fortunately, as the refrigeration could not be secured in time, the air conditioning was abandoned.

An idea just came to my mind in connection with the difficulty encountered in the determination of the exact heat flow with the variation of the volumetric capacity. I should like to ask of the authors if it is possible to simulate the natural conditions by artificial input of the solar effect and the outside surface coefficient. Could one use the plate method or other radiant source of heat and then vary the input cyclically by means of a rheostat and observe what happens to the volumetric absorption or heat storage in the building material.

MR. WRIGHT: With regard to the last question, I assume that Mr. Hadjisky wants to inquire whether the volumetric specific heat could be determined by some kind of experimental setup. Was that it?

MR. HADJISKY: Yes, because you could not depend upon steady solar conditions in the field.

MR. WRIGHT: If a laboratory research program were going to be conducted to determine from the heat flow characteristics of a wall under unsteady conditions, the volumetric specific heat, it would be preferable to consider a little different problem of unsteady heat flow. It would be much simpler to control the temperature of a metallic plate with an electric heater behind it, and to place that heater in contact with the surface of the wall sample. That would be a problem of periodic heat flow through a wall which had a periodic surface temperature which was known. It could be assumed that the surface temperature of the wall followed the variable temperature of this heated plate in contact with it, and then if the heat

flow through the wall were measured by some method, the results would be somewhat better than when trying to vary an air temperature.

One important point that comes up in that connection is that mention inside film coefficient of heat transfer has been omitted. In every case the assumed value of 1½, the rate of heat transfer in Btu per hour per square foot per degree difference in temperature between the surface of the wall and the room air. This was done because measurements of the true natural convection film coefficient of heat transfer show that this film coefficient, the air film coefficient of heat transfer, depends on about the fourth root of the difference in temperature between the wall surface and the air temperature. For normal values of the difference between wall surface temperature and air temperature, the film coefficient of heat transfer for natural convection is about 0.5, which is only one-third of the value that is usually assumed for the so-called inside film coefficient.

The other two-thirds of the value of 1½ comes from the transfer of energy to other solid surfaces in the room by radiation. To be added to the 0.5 for natural convection, therefore, is the value of 1, which is an equivalent film transfer coefficient which takes radiation into account.

In the case of radiant heat transfer, the wall surface may *see* many other surfaces that are approximately at room temperature, such as inside partitions, furniture, etc. But in a room with two outside walls, the wall under consideration also *sees* another outside wall which may be at a temperature higher or lower than the air temperature so that the value of 1, which is added to the 0.5 for true air film convection, is exceedingly variable.

Referring to the experimental problem, it would be difficult to determine accurately in that case just what would be the rate of heat transfer from the surfaces, that is, determining the so-called film coefficients of heat transfer, and that is the reason why it is recommended that, instead of using the varying air temperature, the varying plate temperature in contact with the wall be used.

The mathematical solution for that problem could be very easily carried forward, and results calculated in that way. It is highly desirable that at some time this be done in order to check against the values of the volumetric specific heat measured on another basis.

It has been shown that in some cases the measurement of the insulating value of the material will differ according to the method in which the experiment is carried out, and it is highly desirable that both experiments be carried forward in this case.

H. B. NOTTAGE, East Hartford, Conn.: In regard to the problem of employing some simple experiment to determine the volumetric specific heat, or, what is even of more direct application to periodic or transient heat flow problems, the *thermal diffusivity* (ratio of thermal conductivity to volumetric specific heat), I would like to offer a bit of assurance that the details are indeed quite simple. I have seen it done with the aid of a setup which could be placed in a small box on a table and a technique which was quite elementary and yet sufficiently precise.

It is not necessary to impose an actual periodic temperature fluctuation; the simpler transient case of heating a slab, initially at a uniform temperature, with one face suddenly changed to some other constant temperature is much more easily managed.

The arrangement I have in mind was set up as a student laboratory experiment by Professor London of Stanford University, Calif. He employed a metal plate set in an ice bath for the purpose of applying a sudden constant temperature disturbance at one face of the specimen which was initially at room temperature. Then, the temperature distribution within the specimen was measured as a function of time through the use of carefully located thermocouples. With these data, a very simple graphical analysis leads to the determination of the thermal diffusivity directly, and either the volumetric specific heat or thermal conductivity follows once the other

is known. Values obtained in this manner, so far as I know, check very well with other sources.

MR. WRIGHT: I am glad that Mr. Nottage brought that up. Incidentally, the physicist often measures the properties of steel bars in the way that was mentioned by Mr. Nottage.

1232

No. 1232

FRICTION HEADS DUE TO WATER FLOW IN COPPER, BRASS AND OTHER SMOOTH PIPES

By F. E. Giesecke,* College Station, Tex.

About 1902 a thorough and extensive study[1] was made at Cornell University of the resistances to the flow of water in pipes and the results of this study were presented at a meeting of the *American Society of Civil Engineers* on September 2, 1903. Among the pipes used by Saph and Schoder in their study were 15 brass pipes ranging in size from 2.090 to 0.0107 in.

In 1913 a German mathematician published his classic study[2] of the friction in fluids and among the experimental data studied by Blasius were those of Saph and Schoder, which he pronounced the most carefully executed and the most extensive available. Of the 15 brass pipes tested by Saph and Schoder, Blasius used 8, ranging in size from 2.090 to 0.0107 in. For each test result he calculated Reynolds' number (R) and the value of f in the formula

$$h = f \frac{l}{d} \frac{v^2}{2g} \qquad (1)$$

and plotted f against R to logarithmic coordinates. From the resulting chart it appeared that a definite relation existed between f and R, and Blasius concluded that this relationship could be expressed, with sufficient accuracy, by the formula

$$f = 0.3164R^{-0.25} \qquad (2)$$

Of the experimental data secured by Saph and Schoder 36 were for brass pipes and used by Blasius in his studies as shown in Table 1 and in Fig. 1. The chart of Fig. 1 is drawn to semi-logarithmic coordinates in order that the values of f may be read easily and accurately. The line representing Blasius' formula, Equation (2) appears as a curved line on this chart instead of as a straight line as it would if the chart had been drawn to logarithmic coordinates.

A complete record,[3] with discussions, of the very valuable hydraulic research conducted by John R. Freeman in the Jackson Co. cotton factory at Nashua, N. H., and completed in 1892, 50 years ago, was published in 1941.

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* Professor Emeritus, Agricultural and Mechanical College of Texas. Member of A.S.H.V.E.

[1] Results of investigations made by A. V. Saph and E. H. Schoder at Cornell University on the resistances to the flow of water in pipes. (Published in *A.S.C.E. Transactions*, Vol. LI, pp. 253-309.)

[2] Friction in Fluids, by Dr. H. Blasius, Hamburg. (Published in *Mitteilungen über Forschungsarbeiten*, Vol. 131, pp. 1-40.)

[3] Flow of Water in Pipes and Pipe Fittings, by John Ripley Freeman, C. E., (edited by Clarke Freeman, son of J. R. Freeman).

Presented at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January 1943.

TABLE 1—TESTS BY A. V. SAPH AND E. H. SCHODER—1903

EXPERIMENT No.	FRICTION HEAD FT PER 100 FT	TEMPERATURE F	VELOCITY FPS	f	R
BRASS PIPE NO. 2; DIAM. = 0.1742 FT = 2.09 IN.					
1	4.155	36.0	4.591	0.0222	44,700
11	4.377	49.5	4.893	0.0205	53,600
3	1.342	36.0	2.412	0.0260	23,600
29	0.892	69.8	2.063	0.0235	33,900
33	0.148	69.8	0.735	0.0308	12,070
716	0.812	38.5	1.825	0.0273	18,620
BRASS PIPE NO. 3; DIAM. = 0.12484 FT = 1.498 IN.					
194	17.72	68.8	8.994	0.01761	104,500
550	19.36	35.5	8.738	0.0204	60,300
39	3.44	69.1	3.568	0.0217	41,400
547	6.49	36.0	4.708	0.0235	32,800
203	0.291	71.8	0.870	0.0309	10,520
549	1.12	37.6	1.713	0.0306	12,310
BRASS PIPE NO. 4; DIAM. = 0.10311 FT = 1.237 IN.					
303	15.43	62.0	7.135	0.02015	62,100
721	14.53	37.2	6.516	0.02275	38,400
344	5.45	54.6	3.89	0.0239	30,500
723	4.99	38.0	3.542	0.0264	20,800
45	1.39	69.0	1.813	0.0281	17,450
724	2.12	38.5	2.167	0.0299	13,050
BRASS PIPE NO. 7; DIAM. = 0.05251 FT = 0.630 IN.					
196	61.10	69.8	9.899	0.0211	48,900
476	67.00	37.8	9.711	0.0240	29,500
53	12.61	70.0	4.028	0.0263	20,000
473	20.47	46.0	4.949	0.0283	17,300
388	0.345	85.8	0.538	0.0403	3,260
56	0.652	70.7	0.728	0.0416	3,650
BRASS PIPE NO. 9; DIAM. = 0.03137 FT = 0.376 IN.					
176	172.4	67.5	12.28	0.0231	35,000
514	170.1	37.7	11.415	0.0263	20,600
518	55.8	37.8	6.015	0.0312	10,900
179	40.9	69.3	5.334	0.0290	15,620
519	27.4	37.9	4.001	0.0345	7.270
284	14.08	67.0	2.903	0.0337	8,250
BRASS PIPE NO. 11; DIAM. = 0.02347 FT = 0.282 IN.					
274	208.6	57.5	10.872	0.0268	20,200
487	199.02	39.1	10.170	0.0291	14,170
483	58.35	37.1	4.975	0.0356	6,670
80	43.99	69.0	4.567	0.0318	10,000
139	8.89	70.0	1.832	0.0401	4,070
80	29.37	69.0	3.626	0.0348	7,950

Among the pipes tested by Freeman were 5 brass pipes, ranging in size from 4 in. to ½ in. and varying in length up to 122 ft. These pipes were made up of pieces 12 to 15 ft in length and carefully joined at the ends with special sleeve couplings made of heavy tinned plate. Mr. Freeman states, *Rarely, if ever, in practical work could straighter lines of pipe be secured.* Of the experimental data secured by Freeman 35 are shown in Table 2 and in Fig. 2. Since the data shown on Fig. 2 range almost up to $R = 1{,}000{,}000$, while those of

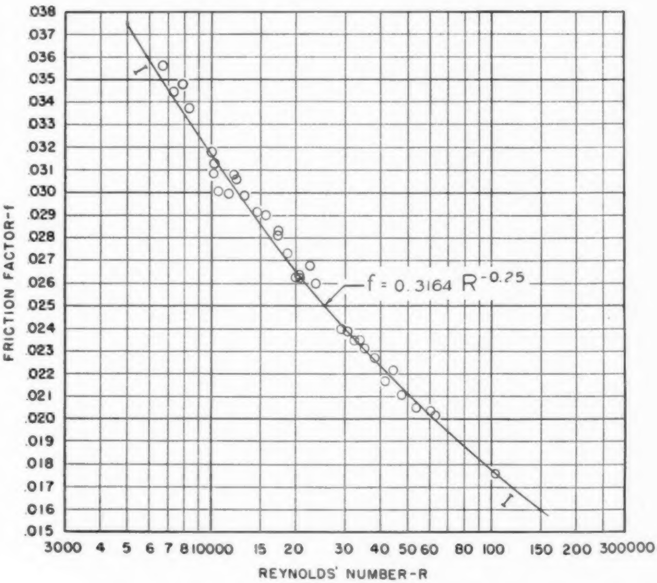

FIG. 1. THE SAPH-SCHODER-BLASIUS *f-R* LINE

Thirty-six of the Saph and Schoder test results and Blasius' *f-R relation:* $f = 0.3164\ R^{-0.25}$ which is probably not accurate for values of R above 75,000.

Fig. 1 do not range beyond $R = 100{,}000$, the character of the relationship between f and R can be determined more accurately from Fig. 2 than from Fig. 1. If Fig. 2 had been drawn to logarithmic coordinates, it would show clearly that the f-R relationship is not rectilinear and that Blasius' formula can be used only for a limited range of R values. In discussing this feature Clarke Freeman states:

> It is our opinion that, had my father's experiments been published even as early as 1900, it is doubtful Blasius would have ever published his straight-line friction-factor equation which has been so widely used and which is the basis of the *one-seventh power law* velocity-distribution equation, valid only over the range $R = 10{,}000$ to $R = 100{,}000$, according to the experiments of J. Nikuradse.

TABLE 2—TESTS BY J. R. FREEMAN—1892 AND EARLIER

Experiment No.	Temperature F	Velocity fps	Friction Head Ft	Friction Head Ft per 100 ft	f	R
BRASS PIPE ½ IN.; AVERAGE INTERNAL DIAMETER 0.5470						
498	68	10.22	49.65	81.44	0.02278	42,710
500	68.5	8.013	63.55	52.09	0.02370	33,680
502	68.5	5.448	32.37	26.53	0.02612	22,900
503	69	4.137	20.39	16.71	0.02853	17,510
504	69	3.064	12.09	9.913	0.03084	12,970
507	70	2.123	6.359	5.212	0.03377	9,107
509	71	1.305	2.695	2.209	0.03792	5,670
BRASS PIPE 1 IN.; AVERAGE INTERNAL DIAMETER 1.061						
541	67.5	17.66	56.44	92.05	0.01686	143,100
542	67.5	15.42	44.81	73.09	0.01756	124,900
543	67.5	13.69	72.03	58.75	0.01789	111,000
544	68	11.79	27.39	44.69	0.01836	96,180
547	68	5.101	12.38	10.09	0.2215	41,620
549	68.5	3.359	6.031	4.919	0.02489	27,600
550	68.5	1.220	0.5219	0.8513	0.03264	10,000
BRASS PIPE 2 IN.; AVERAGE INTERNAL DIAMETER 2.107						
529	66.5	3.760	1.558	2.550	0.02039	59,510
521	69	17.88	50.64	41.51	0.01468	292,600
524	69.5	10.31	18.71	15.34	0.01631	169,800
528	66.5	5.922	6.946	5.694	0.01835	93,670
531	66.5	1.814	0.8499	0.6967	0.02392	28,700
532	67	1.004	0.3115	0.2553	0.02863	15,980
534	67	0.6892	0.1596	0.1308	0.03111	10,980
BRASS PIPE 3 IN.; AVERAGE INTERNAL DIAMETER 3.083						
599	66	30.88	44.84	73.00	0.01258	705,700
603	66	14.08	21.41	17.48	0.01449	322,000
607	66.5	26.31	67.26	54.92	0.01305	605,600
608	66.5	22.02	48.51	39.61	0.01343	506,900
615	67	2.273	0.8122	0.6632	0.02111	52,670
611	66.5	9.092	4.862	7.917	0.01574	209,200
613	67	5.986	5.563	3.725	0.01709	138,700
BRASS PIPE 4 IN.; AVERAGE INTERNAL DIAMETER 3.999						
571	67.5	28.71	57.14	46.78	0.01217	873,700
574	67.5	19.65–	14.31	23.30	0.01294	597,900
577	68	13.74	14.79	12.11	0.01375	421,100
580	69	22.11	35.32	28.91	0.01268	686,700
583	69.5	11.80	5.603	9.223	0.01420	368,900
586	69.5	5.817	3.122	2.555	0.01619	181,900
588	70	1.583	0.3044	0.2492	0.02133	49,820

Studies described in a paper[4] published in 1930 relate primarily to changes in hydraulic characteristics of pipes with service, but they include, necessarily, studies of the friction heads in new pipes. The authors concluded from their

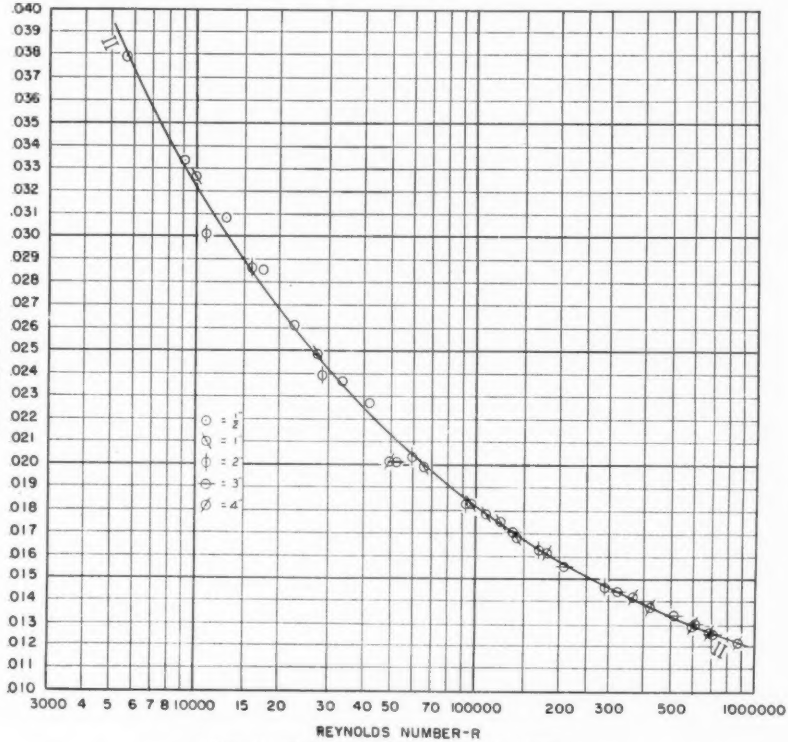

FIG. 2. THE J. R. FREEMAN *f-R* LINE

Thirty-five of Freeman's test results and the line representing the *f-R relation* as determined by Freeman.

studies that for copper, red brass and admiralty metal pipes, 1½ in. and ¾ in. in diameter, the friction heads may be calculated by the formulae:

$$h = 0.307\ d^{-1.246}\ v^{1.754} \text{ ft per 1000 ft and for 59 F water} \quad . \quad . \quad . \quad (3)$$

$$h = 0.247\ d^{-1.246}\ v^{1.754} \text{ ft per 1000 ft of pipe and for 140 F water} \quad . \quad (4)$$

if d is in feet and v in feet per second. In accordance with these equations, four friction heads were calculated for ¾ in. pipe and 59 F water; four for

[4] Hydraulic Service Characteristics of Small Metallic Pipes, by G. M. Fair, M. C. Whipple, and C. Y. Hsiao, Research Fellow, read before *New England Water Works Association*. (Published in *Journal* of *NEWWA*, Vol. XLIV, No. 4, and reprinted as Publication from Harvard Engineering School, No. 58, 1930-31.)

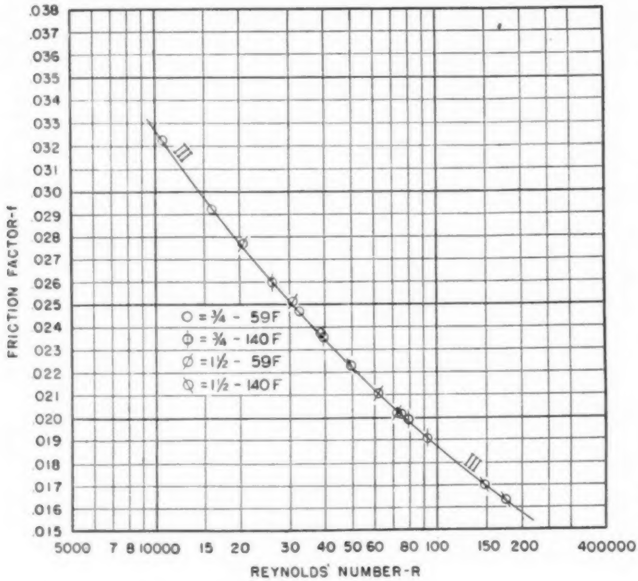

FIG. 3. THE FAIR-WHIPPLE-HSIAO f-R LINE

Sixteen values calculated by means of formulae $h = 0.307\ d^{-1.246}\ v^{1.754}$ and $h = 0.247\ d^{-1.246}\ v^{1.754}$ for 59 F and 140 F water, respectively; and line representing the average f-R relation.

TABLE 3—TESTS BY G. M. FAIR, M. C. WHIPPLE, AND C. Y. HSIAO—1930 AND EARLIER

NOMINAL DIAMETER INCHES	ACTUAL DIAMETER FEET	TEMPERATURE F	VELOCITY FPS	FRICTION HEAD FT PER 100 FT	f	R
¾	0.0676	59	2	2.9719	0.0323	11,000
¾	0.0676	59	3	6.0520	0.0293	16,400
¾	0.0676	59	6	20.413	0.0247	32,900
¾	0.0676	59	7	26.750	0.0238	38,400
1½	0.1273	59	2	1.3506	0.0277	20,600
1½	0.1273	59	3	2.7504	0.0251	30,900
1½	0.1273	59	6	9.2768	0.0211	61,900
1½	0.1273	59	7	12.157	0.0203	72,200
¾	0.0676	140	2	2.3911	0.0260	26,400
¾	0.0676	140	3	4.8692	0.0236	39,600
¾	0.0676	140	6	16.423	0.0199	79,200
¾	0.0676	140	7	21.522	0.0191	92,400
1½	0.1273	140	2	1.0866	0.0223	49,700
1½	0.1273	140	3	2.2129	0.0202	74,600
1½	0.1273	140	6	7.4638	0.0170	149,200
1½	0.1273	140	7	9.7810	0.0164	174,000

¾ in. pipe and 140 F water; four for 1½ in. pipe and 59 F water; and four for 1½ in. pipe and 140 F water. Then, for each case the values of f and R were calculated. The results are shown in Table 3 and in Fig. 3.

In a study[5] conducted at the Agricultural and Mechanical College of Texas in 1931, friction heads were determined for 84 F water flowing in 1¼-in., 1-in., and ½-in. copper tubes 30 ft in length. The investigators concluded that, under these conditions, the friction heads could be calculated with sufficient accuracy by means of the formula:

$$h = 1.16\, d^{-1.22}\, v^{1.7} \text{ milinches per foot for 84 F water} \quad . \quad . \quad . \quad . \quad (5)$$

if d is in inches and v, in feet per second. Equation (5) becomes:

$$h = 0.319\, d^{-122}\, v^{1.7} \text{ ft per 1000 ft for 84 F water} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

if d is in feet and v, in feet per second. Five friction heads were calculated by means of Equation (5) for each of the four pipe sizes and then, for each case, the values of f and R were calculated. The results are shown in Table 4 and in Fig. 4.

Comparing Fig. 4 with Fig. 3 and Equation (6) with Equation (4), it appears that the two equations are similar but that in Equation (4) the exponents of v and d were determined to a higher degree of accuracy than was the case in Equation (6). In Fig. 3 the 16 calculated values lie practically on an unbroken line; whereas, in Fig. 4 the five calculated values for the four pipe sizes lie on four distinct lines so that for any one value of R the extreme f values vary about 3 per cent from the mean. The explanation may be that, while Equation (5) is sufficiently accurate for ordinary design purposes, it is not sufficiently accurate for the study in hand.

The four f-R lines shown in Figs. 1, 2, 3, and 4 were transferred and reproduced in Fig. 5. It is evident from this figure:

1. That the Freeman f-R line extends over a wider range of R values than any of the other three lines.

2. That the Freeman f-R line shows values which differ only slightly from those shown by the other lines. For example, for an R value of 40,000, which is near the center of range of the other three f-R lines, the value shown by the Freeman line is only about 2 per cent lower than the average of the values shown by the other three.

It seems, therefore, that the Freeman f-R line should be used as the basis for friction head calculations for hydraulically *smooth* pipes, especially, also, since the Freeman values are based on tests of pipe ranging in size up to 4 in.; whereas, the other lines are based on tests of pipe ranging up in size only to 2 in.

Before deciding definitely on an f-R line for friction head calculations, it should be borne in mind:

1. That the Freeman tests were made with very long pipes. Of the 35 tests recorded in Table 2 and Fig. 2, 25 were longer than 100 ft.

2. That the friction head in a pipe is the result of the friction between the molecules of the water in the liquid stream. During turbulent flow—which exists for all velocities corresponding to R values above 3,000, approximately—a turbulence is produced in the liquid stream, possibly by the roughness of the inner pipe surface, and other

[5] A.S.H.V.E. RESEARCH REPORT No. 99—Loss of Head in Copper Pipe and Fittings, by F. E. Giesecke and W. H. Badgett. (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.)

factors. This turbulence attains a steady state in a long straight pipe and in turn produces a steady or constant f value. When the water flows through an elbow or valve, or any other similar obstruction, an additional turbulence is produced. This additional turbulence continues for a considerable distance beyond the obstruction in which it was produced and is superimposed on the turbulence which exists as the result of ordinary turbulent flow. Consequently, when water is flowing through a pipe fitting into a long straight pipe, the average friction head in that pipe becomes less and less as the pipe length becomes greater and greater. The friction factor will be less for a pipe 50 ft long than for a pipe 20 ft long.

Since, in ordinary hot water heating systems, the lengths of straight pipe runs between fittings are generally less than 12 ft and, since the Freeman

TABLE 4—TESTS BY F. E. GIESECKE AND W. H. BADGETT—1931–32

Nominal Diameter Inches	Actual Diameter Feet	Temperature F	Velocity fps	Friction Head Ft per 100 Ft	f	R
¾	0.0676	84	2	2.771	0.03015	15,240
¾	0.0676	84	3	5.521	0.02671	22,860
¾	0.0676	84	5	13.149	0.02290	38,100
¾	0.0676	84	6	17.937	0.02169	45,720
¾	0.0676	84	7	23.310	0.02071	53,340
1	0.0879	84	2	2.010	0.02845	19,820
1	0.0879	84	3	4.0052	0.02519	29,740
1	0.0879	84	5	9.545	0.02161	49,560
1	0.0879	84	6	13.013	0.02046	59,470
1	0.0879	84	7	16.912	0.01954	69,380
1¼	0.1076	84	2	1.57	0.02720	24,260
1¼	0.1076	84	3	3.13	0.02410	38,380
1¼	0.1076	84	5	7.46	0.02068	60,640
1¼	0.1076	84	6	10.17	0.01958	72,770
1¼	0.1076	84	7	13.22	0.01870	84,900
1½	0.1273	84	2	1.28	0.02623	28,700
1½	0.1273	84	3	2.55	0.02324	43,050
1½	0.1273	84	5	6.08	0.01994	71,750
1½	0.1273	84	6	8.29	0.01888	86,080
1½	0.1273	84	7	10.77	0.01802	100,420

results are based on much longer pipes, it is wise to add a certain percentage to the Freeman values if the resulting friction heads are to be used in the design of ordinary hot water heating systems.

For these reasons the heavy line in Fig. 5 was drawn to show f values 5 per cent higher than those shown by the Freeman line, and this line is suggested as the basis for friction head calculations for the design of hot water heating systems. Having such a line, it is easy to calculate the friction head for any desired condition if the R value for that condition is known. To find the R value it is necessary to know the kinematic viscosity of the water for the desired condition.

The author was unable to find a satisfactory table of kinematic viscosities in English units, so he prepared Table 5. The values in this table are based on data found in International Critical Tables and other reference books.

Having the *f-R* line of Fig. 5 and the kinematic viscosities of Table 5, the friction heads may be found for any condition. For example, for a 4 in. Type *M* copper tube, when water at 200 F is flowing in the pipe at a velocity of 8 fps, Reynolds' number (*R*) is found by multiplying the velocity of the water, in feet per second, by the internal diameter of the pipe, in feet, and dividing the resulting product by the kinematic viscosity in feet squared per second. In the present case *R* is 8 x 0.328 ÷ 0.00000343 or 765,000. For this

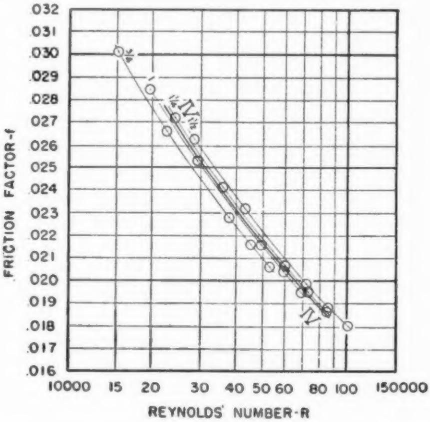

FIG. 4. THE GIESECKE-BADGETT *f-R* LINE

Five values each for 1½ in., 1¼ in., 1 in., and ¾ in. copper tubes and 84 F water calculated by means of the formula $h = 1.16\ d^{-1.23}\ v^{1.7}$, with lines showing the *f-R* relation for the four pipe sizes individually and collectively within the range from $R = 22{,}000$ to $R = 88{,}000$, in which the experimental determinations were made.

value of *R*, Fig. 5, *Curve V*, shows 0.0131 as the corresponding *f* value. The friction is then [Equation (1)] $\dfrac{0.0131}{0.328} \times \dfrac{64}{64.4}$ or 0.003969 ft per foot or 476 mi per foot. If, in this case, the temperature of the water were reduced from 200 F to 140 F, the kinematic viscosity would be increased from 0.00000343 to 0.00000512; the value of *R* reduced from 765,000 to 513,000; the value of *f* increased from 0.0131 to 0.014; and the friction head increased from 476 mi per foot to 509 mi per foot, an increase of about 7 per cent.

To prepare a chart like that shown in Fig. 6 it is only necessary to assume a basic temperature, calculate the friction heads for 2 pipe sizes and for 2 or 3 velocities in those pipes, plot the resulting values to logarithmic coordinates, find the equations of the resulting lines, and combine these equations into one general equation from which the friction heads for the remaining pipe sizes can be determined. For example, if 140 F is selected as the basic temperature

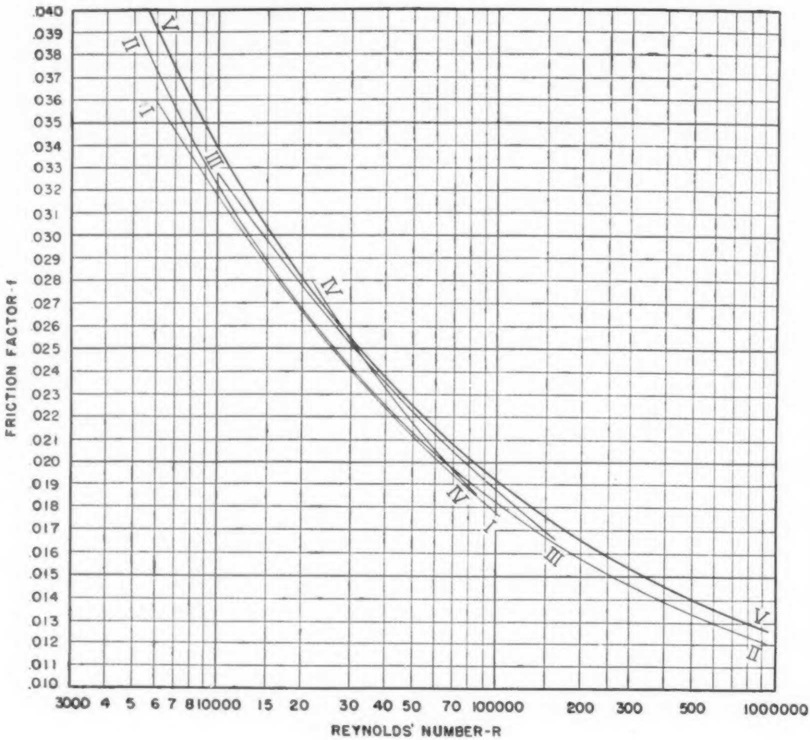

FIG. 5. A COMPARATIVE PRESENTATION OF FOUR f-R LINES OF FIGS. 1, 2, 3, AND 4 AND MAJOR LINE SHOWING VALUES OF f WHICH ARE 5 PER CENT HIGHER THAN THOSE SHOWN BY THE FREEMAN f-R LINE

and friction heads calculated for velocities of 1, 3, and 8 fps in 1 in. and 6 in. pipes, the friction heads will be as shown in Table 6. After plotting these 6 values to logarithmic coordinates as indicated by circles in Fig. 6, the equations for the friction heads may be found to be:

$$\text{For 1 in. pipe, } h = 61.833\, v^{1.7628} \text{ mi.} \qquad (7)$$

$$\text{For 6 in. pipe, } h = 7.346\, v^{1.8102} \text{ mi} \qquad (9)$$

if v is in feet per second.

From these two equations, the general equation,

$$h = 66.07\, d^{-1.2398}\, v^{1.7613\, d^{0.01544}} \qquad (10)$$

if v is in feet per second and d, in in., may be determined and the chart completed as shown in Fig. 6.

Table 5—Kinematic Viscosities of Water

(Feet squared divided by seconds)

F		F		F	
35	0.00001822	105	0.00000700	175	0.00000395
40	0.00001664	110	0.00000665	180	0.00000382
45	0.00001528	115	0.00000633	185	0.00000371
50	0.00001410	120	0.00000604	190	0.00000361
55	0.00001307	125	0.00000578	195	0.00000352
60	0.00001216	130	0.00000554	200	0.00000343
65	0.00001134	135	0.00000532	205	0.00000334
70	0.00001059	140	0.00000512	210	0.00000325
75	0.00000991	145	0.00000493	215	0.00000316
80	0.00000931	150	0.00000475	220	0.00000307
85	0.00000876	155	0.00000457	225	0.00000299
90	0.00000826	160	0.00000440	230	0.00000291
95	0.00000781	165	0.00000424	235	0.00000283
100	0.00000739	170	0.00000409	240	0.00000276

Although the values shown in this chart are not scientifically exact since the construction of the chart is based on a rectilinear *f-R* relation, they are sufficiently accurate to be used in the design of hot water heating systems and in all other ordinary hydraulic calculations relating to 140 F water flowing in copper tubes or other similar pipes. This is evident from Table 7, in which accurately calculated friction heads are compared with corresponding friction heads shown by the chart.

Table 6—Friction Heads Determined from *f-R* Line, Fig. 5, *Curve V*
(For Type M Copper Tubes and 140 *F Water)*

Nominal Diameter Inches	Actual Diameter Inches	Velocity fps	Friction Head mi per ft	f	R
1	1.055	1	61.83	0.0292	17,200
1	1.055	3	422.69	0.0222	51,500
1	1.055	8	2421.72	0.0179	137,300
6	5.881	1	7.35	0.0193	95,700
6	5.881	3	52.82	0.0154	287,200
6	5.881	8	316.82	0.0130	765,800

Table 7—Comparison of Friction Heads Obtained from Chart, Fig. 6, with Computed Values

Nominal Diameter Inches	Actual Diameter Inches	Velocity fps	R	f	Friction Head Calculated	Friction Head from Chart
⅜	0.450	0.8	5,860	0.0396	126	123
2	2.009	4	131,080	0.0181	322	327
8	7.785	7	887,030	0.0128	180	178

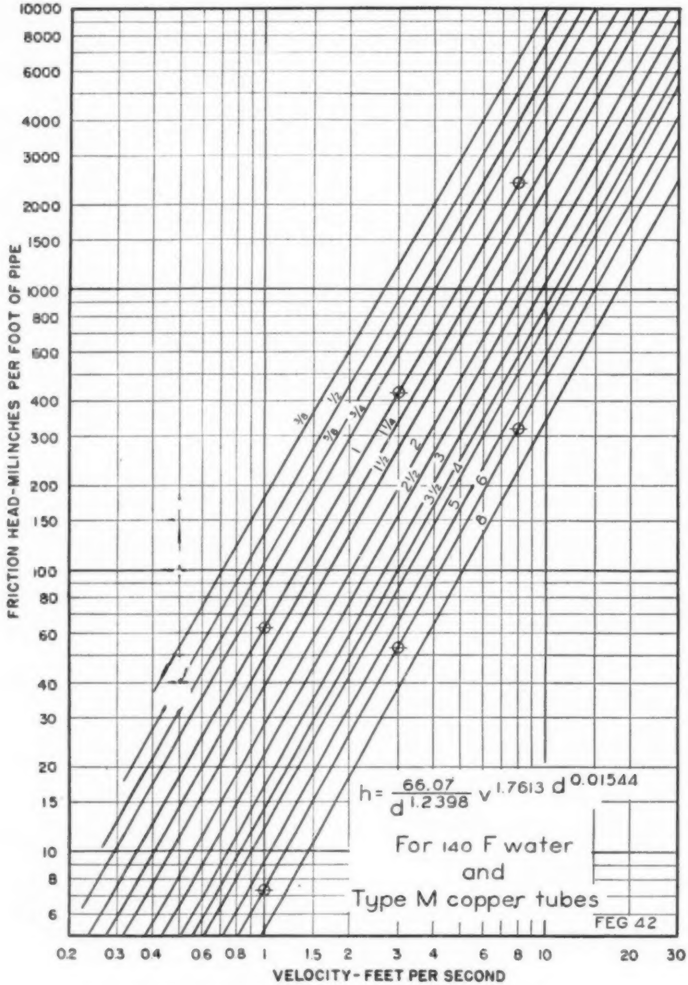

FIG. 6. FRICTION HEADS BASED ON THE MAJOR *f*-*R* LINE

Curve V of Fig. 5 for 140 F water flowing in Type *M* copper tubes. Whenever the temperature of the water differs materially from 140 F or when the water is flowing in iron or other similar pipes, the friction heads shown in this chart should not be used.

The reason for selecting 140 F as the basic temperature is that it is believed to be the temperature at which hot water heating systems operate during the larger part of the heating season. If other temperatures are preferred as basic temperatures, corresponding charts can be easily prepared.

ACKNOWLEDGMENT

The author is indebted to Cadets J. F. Gordon, A. S. Ware, Jr., and R. F. Worth for assistance in preparing calculations and drawings and to Mrs. Barbara Redding for assistance in preparing manuscript and tables.

DISCUSSION

J. N. HADJISKY, Birmingham, Mich.: In regard to the temperature effect, is the author acquainted with some published data which originated during the last war in cooling airplane radiator work in which the variation in temperature was based on velocities of the mass of water or mass of air, and it gave much more interesting results?

One more question is in regard to the smooth pipe *vs.* rough pipe. Could the problem solution be based on momentum and loss of momentum rather than on more complicated figures of diameters and so on?

Given a certain momentum certain surfaces absorb that momentum at a uniform rate for a very smooth pipe or at a variable rate for fittings and very rough pipes. Could it not be determined that for a uniform pipe the resistance is proportional directly to the surface, made up of the diameter times 3.14 times length, which uses up the momentum?

DR. GIESECKE: I am not prepared to answer either question. My investigations have been confined to the details explained.

H. B. NOTTAGE, East Hartford, Conn.: In regard to the temperature effect alluded to by Dr. Giesecke and commented on by Mr. Hadjisky, I would like to offer the problem as an important one for consideration by those who are concerned with superimposed pressure drop and heat transfer design problems. Data on the flow of liquids with superimposed heat transfer in the rather ideal case of relatively long straight pipes indicate an interrrelation of these phenomena which may be treated analytically. However, certain data on air appear to be contradictory; and in the case of many practical systems with flow through passages having a low length to diameter ratio, the exact effect of heat transfer upon the friction factor cannot, to the best of my present knowledge, be predicted accurately by any simple means.

Mr. Hadjisky mentioned some work on airplane radiators just after the last war. I have been concerned with some of the later work on that problem. We still are not able to answer conclusively all questions involving the effect of temperature level and heat transfer upon friction energy loss. The viscosity effect is only part of the story, and surface roughness as well as entrance and exit conditions enter into consideration.

These comments are merely offered more or less in passing for the benefit of others who are interested in similar problems.

DR. GIESECKE: All I can say is that the entire field is very important and very, very complicated, and I believe, deserves considerable additional study.

No. 1233

PHYSIOLOGICAL REACTIONS APPLICABLE TO WORKERS IN HOT INDUSTRIES

By F. C. Houghten, ScD.,* Washington, D. C., Carl Gutberlet ** and M. B. Ferderber, M.D.,† Pittsburgh, Pa.

The past decade has seen a growing interest in the study of industrial hygiene problems in many industries. Much of this interest has dealt with the atmospheric environment. In 1938, under the Technical Advisory Committee on Air Conditioning in Industry,[1] a research project was initiated by the Research Laboratory of the American Society of Heating and Ventilating Engineers to study the relation between the atmospheric environment of the industrial worker and his physiological reactions and degree of comfort.

A comprehensive report (1)[2] of one phase of the study, including a program for continued work, was presented at the January 1939 Meeting of the Society. This paper described the summer season reactions of men engaged in light work to hot atmospheres with 60, 75 and 90 per cent relative humidity. A continued study, reported (2) to the January 1940 Meeting, gave seasonal variations in such reactions, indicating that acclimatization was a factor and that workers have the same reactions at lower effective temperatures during the cool seasons of the year.

Additions to the air conditioning systems serving the psychrometric chambers at the Pittsburgh Laboratory, permitting better control of low humidities, made it possible to extend the study to lower humidities and somewhat higher effective temperatures. The continued study aimed not only to give additional information on this phase of the subject, but also to check further the general assumptions that such reactions are functions of effective temperature.

Test Conditions

Generally, the same test procedure and the same type of subjects were used as in the earlier work. These arrangements and procedures are described at considerable length in the earlier report (1) and only the general outline will be repeated here.

The kind of work performed by the test subjects was designed to simulate many types of employment in modern industry where the worker's activity

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[1] *Committee:* W. L. Fleisher, *Chairman*, L. T. Avery, Lt.-Commdr. A. R. Behnke, U. S. N., Dr. Leonard Greenburg, W. E. Heibel, L. L. Lewis, Maj. W. J. McConnell, (MC), U. S. A., Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, C. Tasker, Capt. R. M. Watt, Jr., U. S. N., and H. E. Ziel.
[2] Numbers in parenthesis refer to appended Bibliography of papers cited.

Presented at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January 1943.

includes little more physical effort than watching an automatic machine and making slight adjustments or corrections, but where keenness of perception, sustained attention and readiness to act are important. To this end, a game

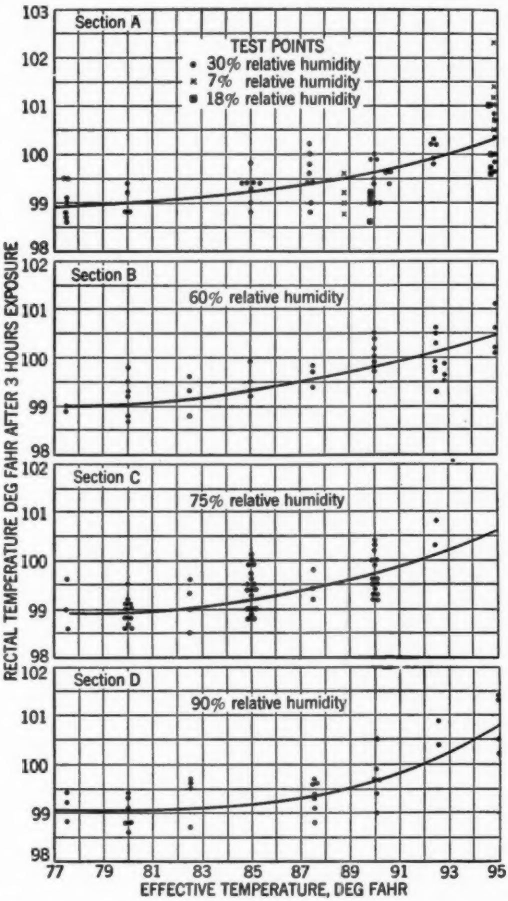

Fig. 1. Relation Between Body Temperature and Effective Temperature of the Environment for Workers in 30, 60, 75 and 90 Per Cent Relative Humidities. A Few Tests at Lower Relative Humidities are also Shown

of chance was introduced that required the close attention of the subjects while they stood or moved about, adjusting small weights as required in the operation of the machine.

Four college students, ranging in age from 18 to 23 years, served as subjects. They were selected carefully after physical and mental examination to determine their fitness for the work and whether their physiological reaction to hot atmospheres, including metabolism, was normal. Only those subjects whose basal metabolism varied little from a previously accepted normal of 40 large calories per square meter of body surface per hour were used.

To determine what constituted normal functioning for each subject, each was seated at rest for 45 min in the comfortable atmosphere of the control room before he entered the test chamber. During this control period and the following 3-hr test period (or a shorter period when the severity of the condition

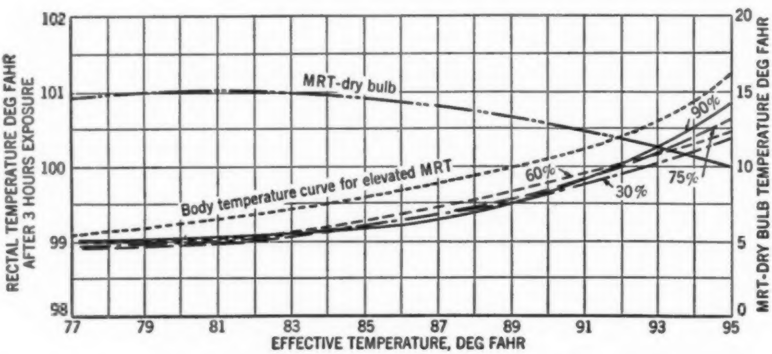

FIG. 2. RELATION BETWEEN RECTAL TEMPERATURE AND EFFECTIVE TEMPERATURE FOR WORKERS EXPOSED FOR 3 HOURS IN VARIOUS EFFECTIVE TEMPERATURES AND HUMIDITIES. BODY TEMPERATURE CURVE RELATIONSHIP FOR ELEVATED MEAN RADIANT TEMPERATURE AND THE ACCOMPANYING INCREASE IN MRT ABOVE DRY-BULB IS ALSO SHOWN

resulted in excessive physiological reaction) the subject's body temperature, pulse rate, blood pressure, leucocyte count, vital capacity, presence or absence of perspiration, and feeling of warmth were determined at frequent intervals. Body temperature was determined rectally.

Tests were made over the effective temperature range from 77.5 to 95 deg, with a relative humidity of approximately 30 per cent. A single test was made at 95 deg ET with the lowest relative humidity that could be maintained by the air conditioning equipment (about 7 per cent) to serve as a check on the physiological reactions at the higher dry-bulb temperature and lower moisture content.

TEST DATA AND RESULTS

The body temperatures of each subject after three hours' exposure in the different conditions at 30 per cent relative humidity are plotted as points in section *A*, Fig. 1; a few points obtained at 18, and 7 per cent are indicated. For comparison, additional data collected in this study, and the results from earlier reports (1), (2), (3), (4) for 60, 75, and 90 per cent relative humidity are plotted similarly in sections *B*, *C* and *D* of Fig. 1.

The body temperature for the different subjects in the various tests shows the usual spread which seems to be inherent in this type of physiological data. The body temperature, however, does show a consistent rise after 3-hr exposure in atmospheres above 85 deg ET, with a measurable rise for conditions above 90 deg ET.

The four curves from Fig. 1 are plotted for comparison in Fig. 2. There is a measurable spread of the curves for different relative humidities above about 90 deg ET, which would seem to indicate that a more pronounced temperature rise may be expected at higher humidities at the same effective temperature. A more careful analysis of the data in Fig. 1, however, seems to contradict this conclusion and emphasizes the probability that this spread is due to insufficient data and the chance falling of the points. In this connection, it is well to point out that the entire spread between the 60 and 90 per cent curves at 95 deg ET could be accounted for by a change of only a little more

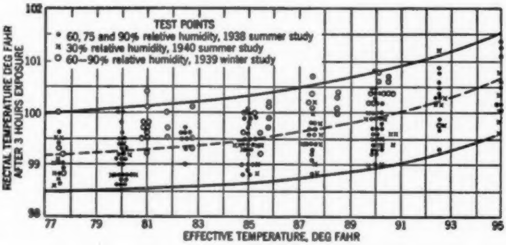

FIG. 3. RELATION BETWEEN RECTAL TEMPERATURE AND EFFECTIVE TEMPERATURE AFTER 3 HOURS EXPOSURE. POINTS GIVEN FOR SEVERAL STUDIES

than one degree effective temperature. Similarly, a change of only 1.6 deg ET (or plus or minus 0.8 deg) would account for the entire spread between the 30 and 90 per cent curves at 95 deg ET.

For comparison, a curve plotted from tests at 60 per cent relative humidity and conditions having a mean radiant temperature higher than the dry-bulb temperature of the atmosphere is also shown in Fig. 2. The data from which this curve is drawn were collected under summer conditions and, therefore, the elevation in body temperature with the presence of radiant heat should be directly comparable with the summer data without radiant heat. The degree to which the mean radiant temperature exceeded the dry-bulb temperature of the air also is given. It should be emphasized that the data available on the effect of radiant heat are so limited that the results should not be considered conclusive; they do, however, add much to the meager present information on this important subject and serve to emphasize the importance of radiant heat in hot industrial environments and the need for a more conclusive study. It is of interest to note that the effect of the elevation of the mean radiant temperature on body temperature is about the same as the effect of one-third this number of degrees elevation in the effective temperature. In other words, about three degrees elevation in the mean radiant temperature seems to have the same effect as one degree elevation in the effective temperature.

All of the data reported in this paper and in the two earlier laboratory reports(1),(2) are plotted in Fig. 3. Those for winter conditions(2) are indicated as circles and show the greater rise in body temperature in winter for the same environmental condition. For all of the variables affecting the data, including winter and summer conditions, variation in relative humidities, in individuals, and for the same individuals at different times, there is a spread of barely two degrees in the body temperature, or a plus or minus spread of less than one degree.

The relative correlation between the elevation in body temperature, and the dry-bulb temperature, the effective temperature index, and the wet-bulb temperature, may be seen by comparing the curves in Figs. 2, 4 and 5. The data plotted in Fig. 4 show conclusively that the dry-bulb temperature is of no value as an index to discomfort and physiological derangements in hot atmospheres. The data shown in Fig. 5 compared with those in Fig. 2 show that the effective

Fig. 4. Relation Between Body Temperature and Dry-Bulb Temperature for 7, 18, 30, 60, 75 and 90 Per Cent Relative Humidity (3 Hour Exposure)

temperature index is measurably better than the wet-bulb temperature index. This is particularly true when the data for 7 and 90 per cent relative humidities are compared in Fig. 5. Although the spread between the 30 and 90 per cent data is not great, it is consistent.

In the earlier study (1) of physiological reactions of workers to hot industrial conditions, the authors, as well as others considering the results, were surprised to find that the subjects engaged in light work while on their feet were comfortable and registered a given degree of moderate perspiration at higher temperatures than had been observed previously (3), (5), (6), (7), (8), (9), (10), (11), for persons seated at rest in still air. This was explained by the authors on the assumption that the workers standing and moving about exposed more of their body surface area for heat dissipation to the atmosphere, and at the same time their movement produced effective air velocity.

The present study offered an excellent opportunity to confirm this fact. Both *at rest* subjects, clothed normally as persons usually are in an audience hall during the summer (*i.e.,* with shirt and light-weight summer coat), and *working* subjects used in this study (without shirt or coat), were exposed to the same atmosphere at 77.5 deg ET. The *at rest* subjects were seated at rest, and the *working* subjects performed their regular tasks. The *working* subjects under this condition were comfortable, while the *at rest* subjects with the

greater amount of clothing and seated at rest registered comfortably warm, and warm.

A similar test was made under the same atmospheric conditions the following day, except that the *at rest* subjects did not wear a shirt or coat. In other words, both subjects were clothed alike. Again, the *working* subjects were comfortable, while three of the *at rest* subjects registered comfortably warm, and one, warm. The *at rest* subjects showed measurably more perspiration. This finding demonstrates clearly that the assumption offered to account for the apparent discrepancy in the earlier study (1) was correct; that is, a person

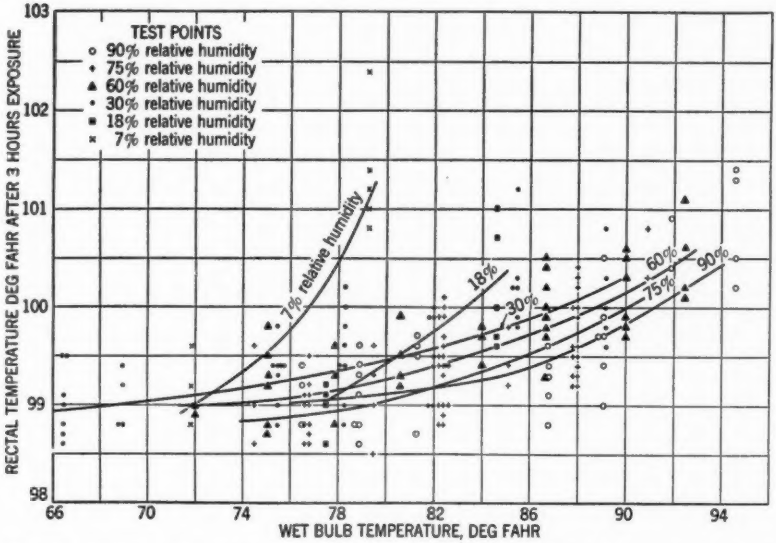

FIG. 5. RELATION BETWEEN BODY TEMPERATURE AND WET-BULB TEMPERATURE FOR 7, 18, 30, 60, 75 AND 90 PER CENT RELATIVE HUMIDITY (3 HOUR EXPOSURE)

may dissipate more easily a slightly greater amount of heat resulting from a higher metabolism when he is on his feet and working than when he is seated at rest.

Vital capacity, blood pressure measurements, and leucocyte counts were made as in earlier studies,(1),(2),(4). However, as these data are less consistent than those for temperature rise and a greater mass of them is required for statistical analysis to show significant variation, there is little advantage in including them here. The same rapid rise in leucocyte count, followed by a fall toward normal when body temperature returned to normal, was shown by the worker when exposed to hot conditions as in the earlier study.

Previous observations(4) revealed that frequent and continued exposures to hot atmospheres resulted in an apparent reduction in the response to automatic

adjustment of the leucocyte count to a higher level with relation to a given hot condition. It is believed that this apparent fatigue of leucocyte control in a worker frequently exposed to hot conditions over an extended period has great significance.

SUMMARY

A study of the rise in body temperature and increase in pulse rate of workers at relative humidities of 30 per cent and lower shows the same reaction at a given effective temperature as previously found for higher relative humidities.

The body temperature rise correlates well with effective temperatures, indicating a possible maximum, plus or minus, departure of less than one degree in effective temperature at 30 per cent relative humidity on a given effective temperature line in the neighborhood of 95 deg ET.

Rather conclusive evidence is reported to substantiate the earlier findings of the Laboratory, that a person is comfortable at a slightly higher effective temperature when standing and moving about at work than when seated at rest.

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DISCUSSION

C. A. MILLS, M.D., Cincinnati, Ohio (WRITTEN): Although Dr. Houghten and his co-workers, as well as others, had previously demonstrated that man does show adaptive changes when exposed to repeated or continuous difficulties in heat loss, they

quite disregard this matter in their present study. No statement is made of the season or prevailing outdoor temperatures when the tests were conducted, nor do they indicate the chronological sequence of the tests made with each subject. Only 4 subjects were used throughout the numerous tests, so that the reactions of each one to the hot environment could well have suffered a considerable adaptive change before the conclusion of the study. Perhaps much of the spread in each group of readings would have been avoided if due attention had been given to this point. In any case, such reports of physiologic reactions should be carefully dated and the sequence of the tests indicated.

As I have often pointed out before, man is highly variable and adaptive in his reactions to changes in the ease or difficulty of body heat loss, and workers entering upon studies in this field should proceed with considerable caution.

W. L. FLEISHER, New York, N. Y.: This paper as presented by Dr. Houghten summarizes to some extent work that has been done by the Society over a period of years; and, it also gives further corroboration to the energy theory which was advanced in a paper[3] presented before the Society in 1941.

One thing that is lacking in this paper but is included in our data relates to the points which indicate the temperatures reached after a three-hour period. A careful analysis of the points which are presented shows that a great many are reached long before the end of the three-hour period, and remain stationary during the entire period.

The reader may get a misconception from the curves of what actually occurs. The curves are meaningless, but the dots have meaning. One thing that is interesting is that the temperature of the subject becomes almost stationary at a definite time. The rise in temperature, or the variations which Dr. Houghten indicated, are due to a great extent to the fact that, as he stated, people have differences in temperature on entering an enclosure. They may be 0.7 of a degree off. They may be as much as 1 deg off. Consequently, those who have lower temperatures show a greater rise in almost every instance. But in the end their temperatures are the same.

The consequence is, that with a 90 deg so-called effective temperature, those people who can stan dthe ambient conditions at all, that is those who are in normal health, take on practically a uniform temperature, not a variation in temperature; and that this uniform temperature, which is higher than the initial temperature at which they entered the enclosure, remains constant, and corresponds almost exactly with the increase in kinetic energy which is coming into them from the higher ambient conditions.

The rise of 1 deg in their body temperature, when multiplied by the mass of the body, will give an energy level which corresponds to the energy level coming to them from outside. Consequently, the further carrying on of this work will indicate, I believe, that the Fleisher kinetic energy theory that was advanced in the previous paper[4] will be upheld as the basic relationship of man to his environment.

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[4] Loc. Cit. Note 3.

No. 1234

THE EFFECT OF CONVECTION IN CEILING INSULATION

By G. B. Wilkes * and L. R. Vianey,** Cambridge, Mass.

Effect of Convection in Ceiling Insulation

IT HAS been well established by many investigators that orientation of air spaces has a marked influence on the rate of heat transfer by convection. It is only natural that one would suspect a similar but lesser effect with loose and light weight insulators where there would be opportunity for free convection. To determine the amount of this effect, if any, a typical guarded box testing equipment was turned to a horizontal position with a test panel on the upper side which would represent a condition for maximum convection.

Test Method

The test panel consisted of a typical frame ceiling of 2 in. x 6 in. joists on 16 in. centers with ⅜ in. plasterboard on the bottom, as shown in Fig. 1. The test area, 32 in. x 32 in., was separated from the guard area by the joists on two sides and by 1 in. x 6 in. boards on the other two sides as diagrammed in Fig. 2, in order to avoid any possibility of convection transfer of heat between the test and guard areas of the panel. Any type of loose fill, bat or blanket insulation could be installed readily in this panel and the coefficient of heat transmittance determined. Shielded thermocouples were used to measure the air temperature on each side of the test panel and surface couples were attached to the lower side of the plasterboard. Fan circulation was maintained in both the guard and test boxes and photoelectric control was used to maintain the temperature of the guard box within a small fraction of a degree of the temperature in the test box.

Thermal equilibrium was established before any test observations were made and the time to complete a test was usually from three to four days. In general the average temperature of the air outside of the panel was about 75 F while the air inside was roughly 120 F, corresponding to a temperature difference of 45 F, which is a normal amount for New England.

The equipment, being new, was checked first with a 3 in. cork slab and the results of this test indicated that suitable results could be expected. The transmittance of the uninsulated ceiling was also determined at this time.

Insulating Materials

Loose Fill Insulation: Rock wool of the *blowing* type from two manufacturers was placed as evenly as possible between the joists on the plasterboard

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in thicknesses varying from 1 in. to 5½ in. A gage board was used to insure uniform distribution and accurate measurement of thickness. The bulk density of the rock wool was kept as low as possible and all samples tested had a bulk density between 5.0 and 6.2 lb per cubic foot.

One test was made with a 2 in. thickness of expanded mica with a bulk density of 7.2 lb per cubic foot.

Bats: Commercial bats from three different manufacturers and of two different thicknesses were tested in the same way. The density of these varied between 2.1 and 3.6 lb per cubic foot.

Blanket Insulation: Insulating blankets from two manufacturers with the thickness varying from 0.5 in. to 2.0 in. were attached to the upper side of the joists, thus making an additional enclosed air space between the blanket and the plasterboard. The blankets were attached to the joists with staples approximately 6 in. apart.

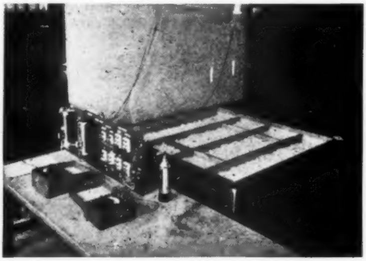

FIG. 1. CEILING TEST APPARATUS

RESULTS

The results are of real importance to the insulation of buildings because in the case of some types of insulation the test values are very much higher than the generally accepted figures used by our engineers and architects.

All values for the coefficients of heat transmission through ceilings in the A.S.H.V.E. Guide, 1942, are calculated with no account taken of whether the heat flow is upward or downward across air spaces, from surfaces to air, or through the insulation. Technical Circular No. 7, Federal Housing Administration, 1940, gives calculated coefficients, for ceilings, in which the direction of heat flow is taken into account for air spaces and surface coefficients. No data were available to indicate the change in the coefficient of the insulating material with the direction of heat flow.

The results of these tests are shown in Table 1, and the accompanying charts show these test values graphically compared with the calculated values based on data in Technical Circular No. 7, Federal Housing Administration.

The test values for loose fill insulation as shown in Fig. 3 average 81 per cent higher than the accepted values today. If the test values for 1 in. and 5½ in. rock wool are omitted because they do not represent common

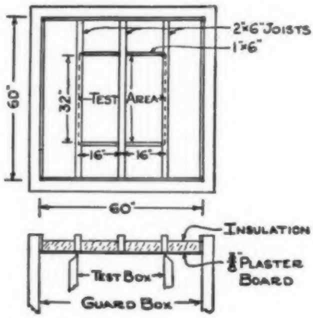

FIG. 2. DIAGRAM OF CEILING TEST PANEL

practice, the remaining tests have coefficients that vary from 75 to 86 per cent greater than calculated values.

The bat type of insulation also gives larger heat transmittance test values than are used in common practice, averaging 34 per cent higher, and individual tests vary from 7 to 61 per cent higher, as illustrated in Fig. 4.

In the case of blanket insulation, one would expect considerable variation due to the nature of the materials and method of installation, since the thickness is uncertain and cannot be duplicated in many cases. From Fig. 5 it is evident that the *U* values for individual tests may vary somewhat, but it is also clear that the average test value for blanket insulation is very close to accepted

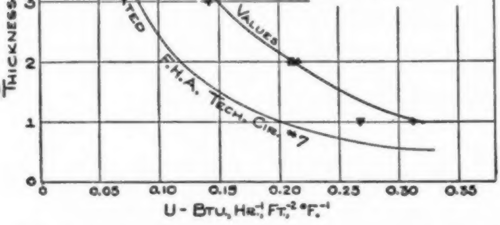

FIG. 3. COEFFICIENT OF HEAT TRANSMISSION FOR VARIOUS THICKNESSES OF LOOSE FILL INSULATION

practice, the difference for the average of these tests being only 2 per cent lower, which is negligible for work of this nature.

DISCUSSION

The 18 tests on ceiling insulation with heat flow upward, covered by this paper, indicate rather conclusively that the present coefficients, used in calculating the rate of heat transfer through ceilings, are subject to very serious

TABLE 1—CEILING TESTS—HEAT FLOW UPWARD

INSULATION		THICKNESS (INCHES)	DENSITY (LB/CU FT)	TEMP. DIFF. DEG F	TEST U[b]	F. H. A. CALCULATION U[b]
Loose Fill	Blowing Rock Wool....	1.0	5.0	44	0.312	0.202
	Blowing Rock Wool....	2.0	5.0	60	0.216	0.116
	Blowing Rock Wool....	3.0	5.0	64	0.142	0.081
	Blowing Rock Wool....	5.5	6.0	80	0.095	0.046
	Expanded Mica........	2.0	7.2	65	0.210	0.116
	Blowing Rock Wool, paper on top........	1.0	5.0	42	0.268	0.202
	Blowing Rock Wool, rough pine floor......	5.5	6.0	78	0.063	0.044
Bat	Bat A................	1.73	3.6	68	0.140	0.131
	Bat C................	1.75	2.1	51	0.154	0.130
	Bat B................	3.25	3.0	74	0.111	0.075
	Bat C................	3.5	2.1	54	0.113	0.070
Blanket	Blanket A............	0.56	..	48	0.289	0.248
	Blanket A............	0.65	..	61	0.207	0.227
	Blanket A............	0.69	..	52	0.220	0.220
	Blanket A............	1.12[a]	..	50	0.162	0.163
	Blanket B............	1.20	..	54	0.129	0.155
	Blanket B............	1.94	..	64	0.108	0.109
Uninsulated....................		...	..	50	0.77	0.80

[a] Note: 2 blankets, each 0.56-in. thick, close together.
[b] U—Coefficient of Heat Transmittance expressed in Btu, hr^{-1}, ft^{-2}, deg F^{-1} (air to air).

error in the case of loose fill insulation, some error for bat type insulation, but otherwise are essentially correct for blanket insulation and uninsulated ceilings.

It is rather obvious that the discrepancy between these tests and calculated values is primarily due to convection. With loose fill insulation, heat flow upward, the air in contact with the plasterboard becomes warmed and then tends to rise vertically because its density is less than the colder air above the insulation. The loose insulation offers little resistance to the passage of this air and consequently heat is transferred by convection from the plasterboard to the upper air. This convection heat transfer is sufficient to nearly double the calculated value of the ceiling coefficient in some cases.

In Table 1, a test is shown with 5.5 in. of blowing rock wool open on top; this test was repeated with the same thickness of insulation and the addition of a rough pine floor with ⅛ in. cracks laid across the top of the joists. The U value was lowered from 0.095 to 0.063 (34 per cent) by this floor. This reduction in U value is far greater than can be accounted for by the insulating

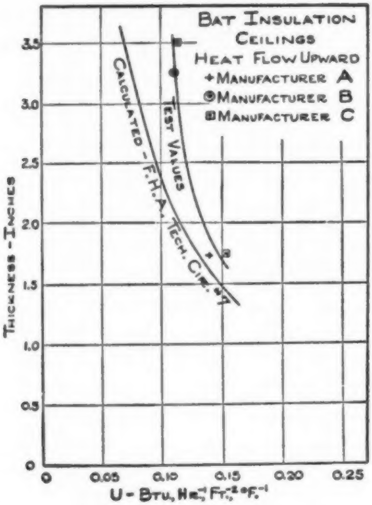

FIG. 4. COEFFICIENTS OF HEAT TRANSMISSION FOR VARIOUS THICKNESSES OF BAT INSULATION

effect of the floor alone. It seems only reasonable to conclude that most of this reduction in value is due to the smaller amount of heat transfer by convection.

An additional check on this effect is shown in Table 1 by comparing the reduction of the U value by laying a single sheet of paper over 1 in. loose rock wool. In this case the U value was reduced from 0.312 to 0.268 (14 per cent) by the addition of one layer of paper. This obviously cannot be

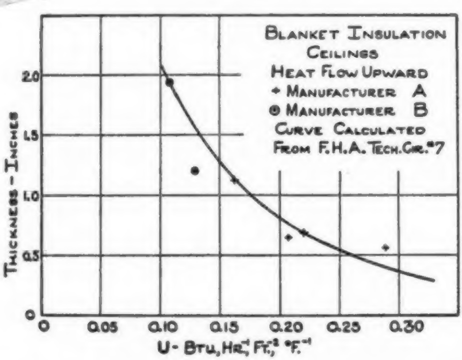

FIG. 5. COEFFICIENTS OF HEAT TRANSMISSION FOR VARIOUS THICKNESSES OF BLANKET INSULATION

TABLE 2—APPARENT COEFFICIENTS FOR ROCK WOOL

THICKNESS INCHES	APPARENT COEFFICIENT, K', FOR BLOWING ROCK WOOL
1	0.53
2	0.56
3	0.52
5.5	0.60

caused by the insulating effect of the paper but is undoubtedly due primarily to the reduced convection.

If one uses the U values as determined by test for the uninsulated and insulated ceilings, the *apparent* coefficient of thermal conductivity, K', of the insulation may be calculated.

Given $U = 0.77$ for the uninsulated ceiling
$U = 0.216$ for the ceiling insulated with 2-in. rock wool

Then $$\frac{1}{0.216} = \frac{1}{0.77} + \frac{2}{K'}$$

And $K' = 0.56$ Btu, hr^{-1}, ft^{-2}, F^{-1} in.

It has been well established by many observers that the K value for rock wool as determined in the guarded plate apparatus is approximately one-half of the value indicated, K'.

Similar calculations were made for the apparent coefficient, K', for other thicknesses of rock wool with the results as shown in Table 2.

The apparent coefficient, K', for blowing rock wool used under the above specified conditions is approximately double the plate test value.

With bat type insulation, the arrangement of the fibers is evidently such that convection is more effectively retarded than with loose fill materials, but there is sufficient convection in some bats to cause considerable error if the usual K value is used for calculations.

The apparent coefficient, K', calculated on the same basis as the loose rock wool above, varies considerably for some manufacturers as shown in Table 3.

The apparent coefficient, K', indicates values from 11 per cent to 70 per cent greater than the generally accepted coefficient, K, for materials of this nature.

The transfer of heat by convection through the insulating medium is effectively stopped by blanket insulation and the test values agree fairly well with

TABLE 3—APPARENT COEFFICIENTS FOR BAT INSULATION

MANUFACTURER	THICKNESS INCHES	APPARENT COEFFICIENT, K'
A	1.73	0.30
C	1.75	0.34
B	3.25	0.42
C	3.5	0.46

the calculated values. These tests give additional confirmation that convection is responsible for the high values in the case of loose fill insulation.

CONCLUSIONS

With heat flow upward through ceiling insulation, the accepted calculated values for U are much too low for loose fill insulation and somewhat low for bat insulation. Transfer of heat by convection in the insulation is responsible for this error. The test values of U for ceilings insulated with blanket insulation appear to check fairly well with present calculations.

For the calculation of heat loss through ceilings, with no flooring and heat flowing upward, the K value for *blowing* type rock wool and expanded mica used as loose fill insulation should be increased about 80 per cent to give results that agree with test values. Other loose fill insulations should also have their K value increased proportionally until further data become available.

In calculating the rate of heat loss for bath insulation under similar conditions, the K value should be increased. The actual amount of this increase varies between 7 per cent and 61 per cent for the three types of bats used in these tests.

No change in the K value for blanket insulation is necessary to make the calculated values agree with these test values.

For the multitude of hómes in the United States, with loose fill insulation in their ceilings and no attic floor, the rate of heat transmission upward through the ceiling can be materially reduced at little or no expense by merely placing a sheet of paper over the insulation.

DISCUSSION

H. E. LEWIS, Toledo, Ohio (WRITTEN) : The results of these tests are very interesting in that they point out the need for comprehensive and continuous research even on materials such as mineral wools, that is, rock, slag and glass fibers which are well recognized for their performance in the home insulation field.

It is believed timely to mention that the *National Mineral Wool Association* undertook rather exhaustive thermal insulation studies sometime ago, under the direction of Prof. F. B. Rowley, University of Minnesota. This research program includes studies of the following phases of insulation, with particular reference to k factors and/or over-all heat transmission coefficients:

1. The effect of installed densities for a given material.
2. The effect of different forms of the same material, that is, bats, blankets, loose wool or pneumatically applied blowing wool.
3. The effect of different insulation thicknesses as installed.
4. The effect of convection currents in ventilated attics, with and without insulation and/or flooring.
5. The effect of position, *i.e.*, whether insulation is installed horizontally or vertically.
6. The effect of different mean temperatures.
7. Comparative results obtained with different types of test equipment, such as guarded hot box *versus* a heat meter, or hot plate.

This research program would have been completed in time for this meeting except for delays due to priority on certain instruments. Without having a copy of the

complete program at hand, it is not possible to state that these objectives as listed are in their entirety; however, they do cover many points for which the *National Mineral Wool Association* has been convinced of the need of more actual test data, to confirm the existing values specified in the A.S.H.V.E. Guide. The experience from a practical standpoint is that the values in The Guide are substantially correct.

Having participated in this program as a member of the Technical Committee of the *National Mineral Wool Association,* it should be mentioned that the results of the tests described in this paper differ considerably from Professor Rowley's findings up to the present time. This statement is based on a recent progress report from Professor Rowley which said in effect, that actual K factors tested to date for mineral wools, are in agreement with values currently given in The Guide.

It is to be expected that there will be *some* range of coefficients reported in such parallel research activities. Such differences may be due to the testing equipment, selection of specimens, test procedures, or to the method of installation used. To avoid extreme variations as in this case it would seem essential to first establish standard equipment and test procedures for guarded hot box tests. Then, with regard to installation methods, the manufacturers' specifications should be followed. For example, blowing wool—as one type of granulated wool is commonly called—should be applied in the test panel by means of recommended pneumatic equipment, not by hand packing.

These comments are made in a spirit of cooperation, not in an effort to advocate one set of data in favor of another. As engineers charged with a responsibility, we want to have the complete facts.

It is to be expected that the comprehensive tests now being conducted as previously described, will contain definite and helpful information on the points covered in this paper. They should be awaited before any decision or acceptance is made.

R. H. Heilman, Pittsburgh, Pa. (Written): The information given in this paper is so radically different from what is commonly believed to be the case, that a rather lengthy discussion is warranted, also a carefully planned series of tests should be conducted before some of the information presented in this paper is fully accepted.

The authors state that the exceptionally high K' in a horizontal position with heat flow upward was primarily due to convection in the insulation. The writer disagrees entirely with this statement as the size of the air spaces in rock wool bats are so minute that no convection currents could be set up in them.

Tests conducted at Mellon Institute about 15 years ago on corrugated asbestos coverings having corrugations slightly greater than 0.1 in. indicated that there were no convection currents in the pore spaces. The test apparatus was set up in such a way as to eliminate all flow of heat upward with free flow downwards. In this case there could be no heat transmitted through the insulation by convection. The flow of heat was then reversed allowing the heat to flow upward through the insulation. Heat loss curves plotted from the results of the two tests which were conducted over a considerable temperature range were found to coincide, thus proving that no heat flow took place by convection.

In a second test two four-ply per inch corrugated coverings, one having the corrugations closed every 3 in., the other having the corrugations open the entire length of the section were tested on both horizontal and vertical pipes. These tests indicated that there was no flue action in the covering having the unbroken corrugations extending for the 36-in. length of the covering when placed on a vertical pipe. It is, therefore, apparent that for rough surfaces, such as asbestos paper, the height of the corrugation has no effect on the convection for air spaces up to 0.2 in. in diameter. Perhaps this is due to the fact that the motionless film of air which is believed to adhere to any heated surface is probably greater than 0.1 in. in thickness. This would prevent any air motion up one side of the insulation and down the other.

In a test conducted last spring a rock wool bat weighing 4 lb per cu ft was cut down to 1.22 in. thickness and applied in a special horizontal set-up on a conductimeter and the conductivity with the top surface exposed to the atmosphere

was determined. This test was only on a small sample and could not simulate the authors' tests in which 6 in. joists were used to simulate regular ceiling construction. However, in the writer's set-up the heat flow was entirely upward and the top surface was free to the atmosphere and in this respect the authors' tests were simulated. After the first test was completed a sheet of asbestos paper was placed over the top surface of the bat and the test was run again. There was no practical difference in the conductivity of the two tests, and the average conductivity obtained

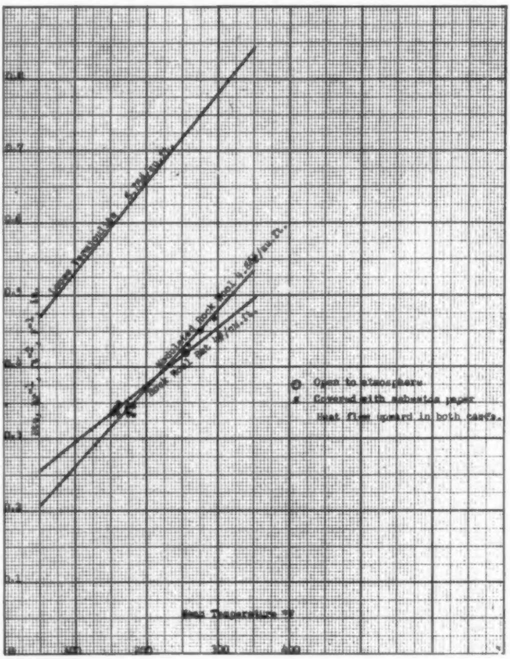

FIG. A. RESULTS OF CONDUCTIVITY TESTS

was 0.272 at 70 deg mean temperature and 0.455 at 300 deg mean temperature. This checks well with the conductivity obtained by the regular method for rock wool bats. A similar test was run using nodulated rock wool which was applied as lightly as possible to the horizontal tester; the resulting density being 4.86 lb per cu ft which is lighter than would ordinarily occur in the hand method of applying rock wool to a horizontal ceiling and which was also lighter than that tested by the authors. A conductivity of 0.228 at 70 deg mean temperature and 0.480 at 300 deg mean temperature was obtained for this set-up. This is actually lower than usually obtained for nodulated rock wool by the regular method of test.

Without changing anything except laying a sheet of asbestos paper on the top of the nodulated wool the test was rerun and no measurable difference in the

conductivity with the rock wool exposed to the atmosphere or when it was covered with a sheet of asbestos paper could be detected.

These tests indicate conclusively that there are no convection currents set up within either nodulated rock wool or bats that would cause the large difference in conductivity as obtained by the authors. The results of these tests are shown in Fig. A.

The authors compare their test values with calculated values based on data in Technical Circular No. 7, Federal Housing Administration, instead of the higher and more nearly correct values as given in the A.S.H.V.E. GUIDE, 1942. The authors make no allowance for the increase in heat flow through the wood joists, nor does the F.H.A. Circular No. 7. The A.S.H.V.E. GUIDE makes a liberal allowance for this factor which is considerable for thick insulation having a low K factor. Obviously, the effect of the joists becomes increasingly less as the thickness of insulation is decreased.

In Table 2 the apparent coefficient K' for 5.5 in. thickness of rock wool is given as 0.60. If we analyze this value for the effect of the joists, because the actual depth of the joists was roughly the same as the thickness of the rock wool and less error would result in making the calculations if the thicknesses were the same, and also 0.60 was the highest apparent K' obtained for the rock wool, we have:

Test area = 32 in. × 32 in. = 7.11 sq ft
Area of wood = approximately 0.96 sq ft
Actual area of rock wool = 7.11 − 0.96 = 6.15 sq ft
Assuming a K' of 0.27 for rock wool as given in A.S.H.V.E. GUIDE, 1942.

$$U^{T} \text{ rock wool} = \frac{6.15}{0.5 + \frac{5.5}{0.27} + 0.27 + 0.51} = 0.284 \text{ Btu per hour per degree Fahrenheit}$$

Assuming a K of 0.94 for dry yellow pine[1]

$$U^{T} \text{ wood joists} = \frac{0.96}{0.5 + \frac{5.5}{0.94} + 0.27 + 0.51} = 0.137 \text{ Btu per hour per degree Fahrenheit}$$

$$U \text{ combined} = \frac{0.284 + 0.137}{7.11} = 0.0592 \text{ Btu per sq ft per hour per degree Fahrenheit}$$

$$U = \frac{1}{0.51 + \frac{5.5}{K'} + 0.5 + 0.27} = \frac{1}{1.28 + \frac{5.5}{K'}} = 0.0592$$

$$1.28\,K' + 5.5 = \frac{K'}{0.0592}$$

$$K' = 0.352$$

The apparent K for the rock wool is thus increased from 0.27 to 0.35 or 30 per cent by the heat conducted through the amount of wood used in the actual test area.

If Fig. A, showing the conductivity obtained on 4.86 lb/cu ft nodulated wool, is examined, it is seen that the conductivity at 70 deg mean temperature is 0.227 while at 115 deg mean temperature, which corresponds to 80 deg temperature difference with air at 75 F, the conductivity is 0.277, or an increase of 22 per cent. The increase from 90 F to 115 F is 11 per cent. The slope of the conductivity curve ordinarily would not be changed much for a density of 6 lb/cu ft.

The authors state that the air inside was roughly 120 F, corresponding to a temperature difference of 45 F, which is a normal amount for New England. The average temperature difference for the rock wool tests as indicated in Table 1 was 62 deg, which corresponds to an inside temperature of 137 F.

There is no objection to testing building insulation at these or higher temperatures, but where the values obtained at these higher temperatures are to be compared with values obtained at lower temperatures, it would be more logical for the investigators to obtain a conductivity curve extending over the range of temperatures

[1] The Thermal Conductivity of Wood, J. D. MacLean. (A.S.H.V.E. TRANSACTIONS, Vol. 47, p. 324.)

actually encountered in practice, from which a true comparison could be made. This is especially true of light weight building insulation which as is generally known very often has a steep conductivity curve.

The primary reason for installing heat insulation in a residence is to conserve fuel in the winter when heat losses become excessive. The mean temperatures at which the insulation is operating in the winter time are usually considerably below 70 F, perhaps 45 to 50 F would be a fair average.

When these lower mean temperatures are taken into consideration, it is believed that an average K factor of 0.27 for all types of rock wool should give a factor of safety instead of having to be increased by 80 per cent as recommended by the authors.

The writer has always felt that the A.S.H.V.E. GUIDE should include K factors, for all materials for at least two mean temperatures so an estimate of the heat gain through insulation could be made for summer conditions. There would be no practical error for the low temperatures encountered in building insulation to extrapolate for other mean temperatures between the two published values.

For the majority of residences where cooling is not employed the K factor of the insulation is relatively unimportant for summer conditions as compared to winter conditions. The heat capacity of the insulation for the summer conditions is probably more important than the K factor.

Also, if the heat loss was increased 80 per cent by convection as stated by the authors at the high mean teamperature at which their tests were conducted, there could be no heat loss by convection at these high mean temperatures in ceiling installations because the heat flow would be downward in the summer instead of upward and this would obviously eliminate any heat loss by convection.

The writer feels that the so-called transfer of heat by convection noticed in some light weight fibrous materials by several investigators, is really not convection at all but radiation, and the flate plate method of test would include this as well as the hot box.

It is strange that the heat transfer by convection should increase the apparent K' of a 3.5 in. thick bat 35 per cent over that of a 1.75 in. thick bat of the same density and manufacturer as shown in Table 3, while the apparent K' of a 3 in. thickness of blowing wool was less than the K' of a 1 in. thickness of the same density as shown in Table 2. The size of the void spaces between adjacent nodules of blowing type rock wool are usually much larger than the spaces between fibers in bat type rock wool.

The only other loose fill insulation reported by the authors was expanded mica. This material had a U^b of 0.210 as shown in Table 1. Using the authors method of calculation, this material would have an apparent K' of 0.578 at a mean temperature of approximately 107 F.

Tests conducted at Mellon Institute on expanded mica which was secured from a shipment for insulating a residence gave a conductivity of 0.54 at 107 F mean temperature as shown in Fig. 1.

There is a difference of approximately 7 per cent between the two tests which is close for materials of this type. The A.S.H.V.E. GUIDE gives two values for expanded vermiculite, one of 0.29 from Peebles tests and the other 0.41 from the Bureau of Standards tests. These tests were conducted on lighter material than either the authors or the Mellon Institute tests. The 0.29 value was for a density of 6.32 lb/cu ft and the 0.41 value for a density of 5.2 lb/cu ft.

It is believed that the K factor for expanded vermiculite should be increased over the figures given in THE GUIDE, but not so much as the 80 per cent suggested by the authors.

Well over 100 conductivity tests have been conducted on various types of mineral wool at Mellon Institute and from the results of these tests, an average value of 0.27 as given in THE GUIDE should not be low for winter conditions where the

mean temperatures are below 70 F. For summer conditions where the mean temperatures are much higher, the writer feels that the coefficient should be increased.

In conclusion the writer can see no justification for the statement made by the authors that all loose fill insulations used in ceilings with heat flow upward should have their K values increased by 80 per cent, on the strength of tests conducted on only two types of loose fill insulation, and on the theory that this increase was caused by convection currents within the insulation. For instance, it is inconceivable how any convection currents could be set up in flaked gypsum at a density of 34 lb/cu ft.

T. T. TUCKER, Atlanta, Ga.: Were different types of mechanical or blow-in insulation tried; that is, different methods of handling the different materials?

MR. VIANEY: Only one method of handling was available, and the authors tried to put it in as well as they could, to correspond with commercial practice. No blowing equipment was available, and the rock wool was put in as closely as possible by hand.

MR. TUCKER: As I understand it, the tests given on blowing rock wool were not made on wool installed by the blow-in method?

MR. VIANEY: It could not be installed by the blow-in method at the time, but that does not prove that it is any different in installation from one blown in, as it was put in at the same densities as would be blown in with the blow-in machine. The authors, however, cannot prove that point.

A. P. KRATZ, Urbana, Ill.: Would the author care to express any opinion on what happens in the case of the wool blown into a vertical wall?

MR. VIANEY: The authors have shown in regular box tests that the wool blown into a vertical wall does not live up to calculated values determined from coefficients. If you calculate a wall with blown-in rock wool you would get about 0.07, and the authors never have been able to get this value, in the wall tests. The authors have never been able to get it lower than 0.09 so that convection must exist even in your vertical wall.

C. F. BOESTER, St. Louis, Mo.: Is that when blown in?

MR. VIANEY: Some of them were blown in, although for these tests the walls were actually given to the authors by the manufacturer. Some of them were installed by the authors and the same results were obtained, if that will make that point clearer.

G. L. TUVE, Cleveland, Ohio: It seems to me that there is something in the situation that needs investigation, not the present paper, but the handling of Society standards.

The idea of these coefficients is that they give the over-all coefficient for a wall. It is a very complicated test by which that over-all coefficient is determined. It would seem that anyone reporting over-all coefficients should be required to report the surface coefficient that accompanies the over-all coefficients in all possible cases.

The author's paper stated that they used fans inside their test box. There are many different ways to place fans inside a test box, and all kinds of different results can therefore be obtained. This is just an example where the Society talks about and gets excited about test data when there could just as well be a get-together to discuss the methods a little bit in some kind of committee to standardize them. Then, when results are presented, there would be less question as to whether or not they are comparable.

There should be a standard method, or at least an attempt should be made to find out ways and means whereby there could be a closer agreement on the present method.

AUTHORS' CLOSURE: The criticisms that have been made were expected. The box the authors have set up is the same as the Standard box that was discussed in a previous paper published some time ago. There was a temporary standard adopted in April 1929. The authors' box was new and they made sure that it was tested in several ways before the tests were actually run.

The authors put in a slab of cork over the whole outfit, and the K_t for that cork came out exactly what it should be as tested with the plate test. If the authors' cork test was the same as with the plate test, why should not the other materials check also?

As far as surface coefficients are concerned, they were measured. The installation of fans in the box was studied by Prof. E. R. Queer, who found that whether he had a fan in there or not the actual U value was practically the same. The authors had a fan, and the surface coefficients averaged from 2.26 to 3.76, an average of 3, which gives us a resistance of 0.33. The surface resistance used by the F.H.A. is 0.50. If you have a considerable amount of insulation in your walls or ceiling the resistance of that will be somewhere in the nature of 13, so whether the resistance for our surface is 0.33 or 0.50 it is going to be a matter of little consequence.

Going back to the data, and the difference in test equipment, the A.S.H.V.E. GUIDE gives 0.77 for an uninsulated ceiling. The authors' result showed exactly the same for an uninsulated ceiling. So, if the authors result was exactly the same for uninsulated and for a cork slab, why should it not be the same for rock wool? As far as blowing it in and the authors putting it in by hand, the authors will grant that point. The contractors do not want to blow in two bags of rock wool when they have 2000 to blow in elsewhere. That is just a point that could not be helped, and as Mr. Lewis pointed out, equipment is hard to get, and sometimes impossible to obtain.

The effect of mean temperature on insulators has not been shown in this paper, because the authors think it is of doubtful consequence as they are awaiting more data on it. One of the purposes of this paper was to stir up interest in obtaining the real data and to find out what it is all about.

The authors welcome the discussion presented by Mr. Heilman, as it offers an opportunity to present rather complete answers to various questions and it enables them to give additional evidence that was not mentioned in the paper in support of the results and conclusions from the investigation.

The tests made at Mellon Institute some 15 years ago on corrugated asbestos coverings seem irrelevant to this discussion. In a horizontal position, this insulation consists of small closed air spaces, except at the ends, and no one would suspect any appreciable convection under such a condition. In the case of the vertical tests on this insulation the length of travel for the convected air was 36 in. and the resistance to air flow through a rough tube of about 0.2 in. diameter was evidently too great to permit much convection. The conditions of these tests are so far removed from the author's tests that it is difficult to see that they serve any useful purpose as an argument against the possibility of convection in the tests under discussion.

With reference to the tests made on the conductimeter, Mr. Heilman states that they "indicate conclusively that there are no convection currents set up within either nodulated rock wool or bats that would cause the large difference in conductivity as obtained by the authors." He also mentions that these tests were made only on a small sample and could not simulate the authors' tests. The authors have no data as to the size of the sample, but it is very dangerous to draw conclusions as

to the amount of convection in large samples from the results on small samples. Furthermore, he states that he obtained a K value of 0.228 at 70 deg mean temperature, which is an exceptionally low value for this type of material. To base conclusions as to the possibility of convection in the authors tests on the results of the conductimeter tests with a small sample (size unknown to authors) and indicating an extremely low K value, does not appear particularly convincing.

The authors' tests, as shown in the paper, checked accepted values for corkboard and various blanket insulations as well as uninsulated ceilings. If these tests checked calculated values so well, why should the tests on nodulated rock wool and bats be questioned when made under the same conditions? Convection appears to be the only reasonable explanation for the high values found. Furthermore, a single sheet of paper placed over the top of the loose rock wool immediately reduced the U value materially. If the source of the high value were radiation, the placing of paper on the rock wool would have only a slight effect because the temperature of the paper would be nearly that of the surroundings.

With reference to the calculation of the value, taking into account the effect of the wood joists, if Mr. Heilman had used the K' value that the tests indicated of 0.60, he would have found that the effect of the joists was relatively small. Instead, he arbitrarily assigned a K value of 0.27 to rock wool under these conditions which leads to a very different result. The 0.27 value is the value found from plate tests but it very evidently does not apply to loose rock wool used as ceiling insulation, exposed to the air on top and with heat flow upward.

The question of high mean temperatures has also been raised. The mean temperature for the ceiling tests varied from 96 F to 118 F. It is true that for a mean temperature of 50 F, the true K value for insulation would be less but in these ceiling tests the authors had convection and convection is not dependent upon mean temperature but primarily upon temperature difference. This makes the variation of the U value with mean temperature considerably less than would be the case with pure conduction. The authors agree with Mr. Heilman that it would have been interesting to have made tests at different mean temperatures so as to establish a relation between mean temperature and heat transmittance, but this would have involved considerably more time and expense than seemed warranted. The tests with blanket and corkboard insulation indicate that the mean temperature is not a very important factor in these tests. It might account for a few per cent but the apparent K value for nodulated rock wool is increased nearer 100 per cent.

The authors are at a loss to understand why Mr. Heilman raises the question of summer conditions with heat flow downward through the ceiling. It is definitely stated in the paper, several times, that the results apply only to upward flow of heat (winter conditions) through ceilings and with the insulation exposed to the air above. Obviously, the results of these tests do not apply to summer conditions.

The writer states that over 100 conductivity tests have been conducted on various types of rock rool at Mellon Institute indicating an average value of 0.27. This value is generally accepted for the result of a standard plate test. The type of tests covered by the paper are under entirely different conditions and the results should not be compared with plate test values where the chance for convection to take place is small.

There are a few references to convection in insulating materials that can be found in previously published data.[2] Most of these investigations referred to were made in vertical plate test equipment and results of convection would have been more

[2] Heat Insulators—National Physical Laboratory, by Ezer Griffiths. (Food Investigation—*Special Report No. 35*—British, p. 21, 1929.)
Heat Insulation as Applied to Buildings and Structures, by E. A. Allcut and F. G. Ewens. (University of Toronto—*Bulletin No. 149*, 1937.)
Thermal Conductivity of Insulating Materials, by E. A. Allcut and F. G. Ewens. (*Canadian Journal of Research*, 1939.)
Properties of Heat Insulating Materials, by E. A. Allcut. (University of Toronto—*Bulletin No. 169*, 1941.)

evident if conditions were such as shown in the authors' ceiling tests. In the first investigation evidence of convection currents was found in granulated cork placed in vertical plate tester. In the investigation described in *Bulletin 149* of the University of Toronto, evidence of convection currents was found in rock wool, 10 lb/cu ft, and this article mentions increasing importance of convection where the density is small. In the 1939 report of the *Canadian Journal of Research* it was pointed out that with shredded red wood bark, they found that at the *same mean temperature,* the *K* value increased 83 per cent as the temperature difference was increased from 10 to 100 F. To quote Allcut and Ewens in this investigation, "these differences are difficult to account for unless it is assumed that convection currents exist in the packed material and they may be reduced by increasing the density of the material or by dividing the specimen into smaller cells." Further confirmation of the effect of convection currents in rock wool and other loose insulators was determined in Bulletin 169 of the University of Toronto.

In view of the evidence given, one would expect, and the authors' tests indicate, that convection plays a very important part in nodulated rock wool placed in a ceiling and exposed to the air above.

1235

No. 1235

SEMI-ANNUAL MEETING, 1943

Pittsburgh, Pa.

WITH an attendance of 300 members, guests and ladies, the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS opened at the William Penn Hotel, Pittsburgh, Pa., June 7th. Despite the fact that the program was tailored to meet wartime conditions the interest of Society members was evidenced by the proportion that were in attendance compared with the small number of guests. All four technical sessions were heavily attended and the number of Technical Committee Meetings was an indication of the importance of the wartime semi-annual conferences. The Council held two meetings and regular meetings were also held by the Committee on Research, War Service Committee, Guide Publication Committee and the Nominating Committee.

Pres. M. F. Blankin called the first session of the Semi-Annual Meeting to order in the Urban Room of the William Penn Hotel on Monday morning June 7, and introduced G. G. Waters, president of the Pittsburgh Chapter.

Mr. Waters presented Frank L. Duggan, President of the Chamber of Commerce of Pittsburgh who in his welcoming address spoke of Pittsburgh as *the arsenal of the allies.* He made a brief reference to the historical high lights in Pittsburgh and referred to its industrial production during World War I.

In a brief response, President Blankin expressed the appreciation of the members for the cordial greeting and also the pleasure it gave him to see the fine attendance at the opening session. Mr. Blankin then gave a brief report as President of the Society and said that a great many difficulties had been encountered this year not only because it was a war year but also because our viewpoint and objectives in research and other Society activities had to be revised. He announced appointment of Carl H. Flink, as Technical Secretary of the Society and said that it was expected that an important announcement regarding research activities would be made before the conclusion of the meeting.

President Blankin said that he had done considerable traveling in visiting Chapters during the past three months and had addressed 18 groups. He also found in three other cities that there was a desire for the organization of local chapters, and a great many potential members of the Society. He said that considerable time and effort was being spent on membership work without any special campaign and that the results have been gratifying and he believed that the goal of 400 new members in 1943 would be accomplished. He requested the cooperation of the members in the work of the Membership Committee and suggested that many men were waiting to be asked to join the A.S.H.V.E.

Lester T. Avery, chairman of the Committee on Constitution and By-Laws presented an amendment offered by a petition of Canadian members from

Toronto, Montreal and Winnipeg. He read the amendment to Art. B-XI as follows:

Section 1—When so directed by the Council, the Chairman of the Finance Committee shall invest such portions of any funds of the Society, as determined by the Council, in securities of the United States Government, *in securities of the Government of the Dominion of Canada,* or in securities legal for the investment of funds of savings banks of the State of New York. Not less than one-half of such invested funds shall be in securities of the United States Government. All investments shall be approved by the Council.

He pointed out that the change in the present section was the addition of the words, "in securities of the Government of the Dominion of Canada." Mr. Avery's motion for adoption was seconded by Professor Rowley and by vote of the members present, it was unanimously adopted.

President Blankin then called on Mr. Avery to present a resolution from the members of Northern Ohio Chapter which he read as follows:

At the regular May Meeting of the Northern Ohio Chapter of A.S.H.V.E. held at Cleveland Engineering Society Club Rooms May 10, 1943, the following motion was unanimously passed: The members of the A.S.H.V.E. who constitute the membership of the Northern Ohio Chapter believe there should be a more direct connection and responsibility between the several chapters and the National Society and

The subject should be carefully considered at this time by all members, and

In order to accomplish this purpose Lester T. Avery is instructed to offer the following resolution from the floor at the Semi-Annual Meeting of the Society in Pittsburgh in June 1943, to wit:

Whereas, the National Society members may not belong to a Local Chapter unless they separately apply for admission and pay local chapter dues, and

Whereas, Local Chapters may not admit to membership anyone who is *not* a member of the National Society, and

Whereas, the nominations for National Officers are the responsibility and privilege of the Chapter representatives, and

Whereas, the National Society seeks the help of these several Chapters in securing new members, in carrying on Society activities, in serving as host to annual and semi-annual meetings, and

Whereas, the continued growth and strength of the National Society is directly related to the growth and strength of the several Chapters,

Resolved, that a special committee called the Chapter Relations (Development) Committee shall be appointed by President Blankin which Committee shall consist of three members of Chapters, only one of whom may be a present officer or Council member of the Society,

And this Committee is to investigate the subject of Chapter-Society relationship and report in writing at the next Annual Meeting, in January 1944, its recommendations as to:

1. A plan whereby each member of the National Society becomes a member of a Local Chapter;
2. A plan for the geographical boundaries of each Chapter thus determining its membership on a convenient geographical basis;
3. A schedule of National dues and Chapter dues taking into consideration the possible financial support of the Chapters by the National Society;
4. Such other matters that may be considered pertinent to this subject.

This report is to be considered on the floor of the meeting and if accepted and approved by the majority vote of members present, it shall be referred to the committee on Constitution and By-Laws to prepare amendments if any are required in order to carry out the recommendations, which amendments shall be submitted to the membership for approval as provided in the constitution.

Mr. Avery moved the adoption of the resolution and the appointment of a special committee by the President of the Society. The motion was seconded by T. T. Tucker, Atlanta, Ga.

In reply to questions about the membership Mr. Avery gave some statistics regarding the Society membership—1925—2372; 1926—2712; 1939—3067; 1940—3147; 1941—3320; 1942—3132; 1943—3006. It was his opinion that there should have been an increase rather than a loss during the past four years which was a period great progress in the industry. He stated that approximately 70 per cent of the Society membership also held Chapter Membership and he thought that the best way to increase the strength of the Society is through the establishment of new chapters and the strengthening of existing chapters.

F. C. McIntosh, Pittsburgh, said that the resolution brought several questions to his mind, for example, what happens to a man who is far from a Chapter center? What is the necessity for financial support of the Chapter? Why is it not possible for a Chapter to carry its own financial responsibilities? He expressed the hope that there would be some discussion of these points for the benefit of the proposed Committee as he believed that before any action is taken the opinion of a cross-section of the Society membership would be desirable.

V. J. Jenkinson, Toronto, thought that a plan could be developed which would provide better coordination between the various Chapters and the national organization and that something could be done for the guidance of incoming Chapter Presidents and Secretaries and Committee Chairmen.

President Blankin said that for the information of members, there was a Chapter Relations Committee, whose duties included the operation of the Speakers Bureau for the benefit of the Chapters and the resolution would provide for a Chapter Development Committee to report on the matters which have been brought up.

Mr. Campbell noted that a speaker had requested some activities that would aid Committee Chairmen in the various Chapters, and he stated that on three occasions as Chairman of the Membership Committee of the Society he had been unable to get the names of the local membership chairmen because of his inability to get replies from Chapter Secretaries. This year he had requested the Chapter Delegates to turn the matter over to the Local Membership Chairman and the results have been better. It was his opinion that chapters were not cooperating, rather than the national organization.

Mr. Avery said that no decision should be made on any of the points brought up but it was desired that a committee be appointed to study the various questions involved. He thought that the Committee could aid in correcting the situation described by Mr. Campbell. He also thought that the Committee could outline a plan regarding the necessity for Chapters in places where they would not be an expense to the Society but would contribute to its growth.

Before calling for a vote on the question, President Blankin commented that there seems to be an erroneous impression that the present Officers and Council are not interested in the formation of new Chapters and he wanted to correct it as the formation of three new Chapters is under consideration at the present time. When the vote was taken, the majority favored the adoption of the motion.

President Blankin then introduced First Vice-Pres. S. H. Downs, Kalamazoo, Mich., and Second Vice-Pres. C.-E. A. Winslow, New Haven, Conn., and he asked Mr. Downs to take charge of the meeting during the presentation of the first paper.

Mr. Downs called attention to the innovation on the program which indicated that a time schedule would be followed. He then introduced H. F. Randolph, Utica, N. Y., who presented the paper on the performance of a residential panel heating system which he and J. B. Wallace had prepared.

At the conclusion of a discussion, President Blankin resumed the Chair and the meeting adjourned at 11:45 a.m.

Get-Together Luncheon

A get-together luncheon was held in the Cardinal Room of the William Penn Hotel, at 12:15 p.m. and at the conclusion of the luncheon, G. G. Waters, president of the Pittsburgh Chapter, introduced the head table guests and then presented Howard Coonley, Director, Conservation Division, War Production Board, Washington, D. C., whose subject was Fuel Conservation.

Fuel Conservation

By HOWARD COONLEY,* WASHINGTON, D. C.

The nation faces a more acute fuel crisis than anyone imagined possible a few years ago. There was considerable talk of fuel shortages last winter but the fact that we squeezed through somehow by the *skin of our teeth* led a large percentage of our population to believe that the situation was not so critical then as it was pictured, and may now cause them to discount the prophecies of a very much greater shortage for the coming winter.

This impression must be erased as quickly and as completely as possible. All indications point to the inevitable conclusion that the American people will have to get along with less fuel this winter than they have heretofore thought possible. With oil stocks already down to 27 per cent of normal [1] on the eastern coast, there can be nothing but pessimistic views of the situation expressed in Washington.

The proper thermal environment, as you of all people know, is one of civilization's greatest contributions to mankind. We have become so accustomed to this environment that the lack of it may prove one of the greatest demoralizing factors during the war. The American people will face that fact and that danger this winter.

The War Production Board has under consideration, the Problem of reducing the fuel requirements of the nation in a dozen different ways. The requirements involve such fantastically high figures that even a small percentage of saving becomes an objective of vast magnitude. In the heating field alone, a 1 per cent saving would amount to 1,720,000 barrels of oil,[2]—the equivalent of 7,224 tank cars. A 1 per cent saving of heating coal would amount to approximately 1,800,000 tons [3] or 36,000 car loads. There is a possibility of making many of these 1 per cent savings.

There are many factors involved in this fuel shortage. One of the primary ones is the lack of transportation facilities. The transportation systems have done an

* Director, Conservation Division, War Production Board.
[1] P.A.W. release, May 20, 1943.
[2] Office of Civilian Requirements states that 172,000,000 bbl oil used for heating in 1941. (By telephone.)
[3] Office of Civilian Requirements.

almost impossible job. The magnitude of that job is reflected in the following figures: Before the war there were practically no oil deliveries to the eastern seaboard by rail. Now the railroads are delivering practically all of the oil up to 950,000 barrels a day to the eastern seaboard. Before the war, tankers delivered 1,400,000 barrels per day [4] to this territory; today they are in the Navy. The fact that our railroads have managed this almost complete switch-over on such a gigantic scale has saved the nation from a major disaster.

They have an increasing number of demands placed upon them and provisions must be made to reduce this tremendous fuel transportation problem which they have, up to now, carried. Peculiarly enough, in the field of residential heating, it is on the east coast—which requires the longest transportation—that the bulk of the heating oil is used. Two-thirds of the 3,500,000 [5] dwelling units in the country heated by petroleum products are on the eastern seaboard. There is also a higher proportion of oil heated homes to non oil heated homes on the eastern seaboard than in any other part of the country, for instance 45 per cent in Rhode Island, whereas it is less than 2 per cent in some of the southern states. In the country as a whole, only 10 per cent of the homes are heated with petroleum products.

Information developed from time to time by you and your associations has convinced us that a substantial reduction in these fuel requirements can be made through the intelligent application of your findings, and that is what the War Production Board is contemplating. This is an emergency. What is needed is quick action. We will have to let refinements go by the board in the interest of speed. The theoretical possibilities of fuel conservation reach fantastic proportions when all possibilities are thoroughly explored. Still, one half, or even one third of the theoretical is a goal which would inspire the cooperation of even the most thoughtless individual, if the opportunity were properly presented. Voluntary cooperation should be tried first, since carefully worked out mandatory regulations will require lengthy discussions and careful charting through seas of legal red tape. This cooperation should not be difficult to obtain, especially if the findings of your and other associations are properly presented to the public. Your reports on fuel conservation possibilities provide an inspiring basis for such a program.

For instance, one organization [6] reports that their survey indicated that only 60 per cent of eastern oil heated homes have any weather-stripping, resulting in a loss of almost 200,000,000 gal of oil per season in district No. 1 alone. This amounts to 6 per cent of the fuel oil sold in that district. It would require only a small amount of weather-stripping to save this 200,000,000 gal. In fact, as one company points out, if zinc weather-stripping were used on an average house, 70 lb of zinc would save from 300 to 400 gal of oil, or as much as is saved by a ton of ordinary insulating material. Now, we are not at this time recommending the use of zinc for weather-stripping because it might be entirely impossible to allocate zinc for even such a tremendous return.

Many weather-stripping installations can be made without the use of any critical materials; for example, wood strip and felt will also do an admirable job. That this saving is not fantastic has been more than proved by many research agencies throughout the country ranging from governmental agencies to fuel associations and university research laboratories.[6]

All these estimates indicate that on the average house, from 15 to 20 per cent of the fuel can be saved by means of proper weather-stripping. If these facts are properly presented to the home owner, it is probable that for the benefit of his own pocketbook, if for nothing else, he will proceed to weather-strip either in the home-made fashion or by a more deluxe job done by experts. It is at this point, however, that Industry and Government must cooperate. It is up to one to advise

[4] P.A.W. release.
[5] U. S. Department of Commerce H-13, No. 5, September 9, 1942 *Housing*. Also, No. 3 of same series.
[6] Oil Heating and Merchandising News Fuel Oil and Oil Heat, August 24, 1942.

and convince the home owner that it would be worth his while to make an investment in order to get a saving of two or three times the amount spent within a few years. It is up to the other to convince him to do his work with non-critical materials and thereby contribute his share towards the fuel conservation program.

Another conservation possibility arises in the field of hot water heating. The householder should be impressed with the fact that 10 per cent of our fuel oil is used in homes for heating water. Here substantial economy is possible if home owners cooperate. Homeowners would keep their hot water at 135 deg if they knew that for every 10 deg above that temperature, 8 per cent more fuel is required, and they cannot use such high temperatures of water anyway.

I also note with interest that many publications in your field estimate that the efficiency of fuel utilization of many heating installations can be raised by 20 to 30 per cent through expert analysis and tune-ups. Your findings in this field indicate that vast numbers of heating systems are improperly regulated as to on and off cycles, nozzles and drafts, which, when corrected by persons who *know how,* will stretch out the fuel oil ration card to a comfortable extent.

The coal furnace, now operated by so many of our millions of people, has come to be looked upon as something which will just run by itself. Surveys indicate that in some areas only 5 per cent are satisfactorily adjusted.[7] And yet, as has been so amply demonstrated, properly reconditioned operating and regulating equipment would save most owners 25 per cent of their fuel bills. Of the 11,000,000 tons of anthracite coal used in homes 23 per cent or 2½ million tons could be saved by automatic controls to keep the inside temperature from rising above 70 deg. This means 17 days of mine production and almost 5,000,000 ton miles of transportation, as well as 9 million car miles, and would save 2½ million gallons of gasoline on retail truck deliveries. This refers to anthracite only.[8]

The insulation of homes, of some of our most dilapidated structures, will save as much as 40 per cent of the fuel bill. Many types of insulation are non-critical and can be installed very simply and at a low cost. The possibilities along this line on the average uninsulated house amounts again to about 25 per cent.

Common sense methods of home management by the housewife can also save more than 10 per cent of the fuel bill. The constantly opened doors by children at play; the fresh air addict who sleeps with all the bedroom doors and windows open; the charming fireplace, racing to empty the house of its heat faster than the heating system can put it in; all these are dubious luxuries which we cannot afford this winter. Home management hints on all these points will be broadcast. We shall inform home-owners that if they maintain a 60 deg temperature at night they save 5 per cent or more of their fuel; if they maintain 68 deg instead of 70 deg during the day, they save 7 per cent, and for those who over-heat, we will point out that it takes 3½ per cent more fuel to heat the home at 71 deg than at 70 deg. The home owner will be healthier and wealthier if he takes these hints seriously.[9]

The use of small radiant electric heaters to provide local and concentrated heat in bathrooms and at breakfast tables will save a tremendous amount of fuel. Their use in the spring and fall instead of operating the central heating system, can also do much to cut fuel consumption.

Many of our homes, as well as industrial establishments, have heating systems without any of the automatic controls which add so much to the fuel efficiency of the system. The possibility of allocating more material for the production of heating controls for installation in these systems is being given consideration. The most expert advice is needed on this point to determine whether we can trade the critical materials and labor involved for the 15 to 20 per cent reduction in fuel consumption that such installations frequently return.

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[7] Coal Heat magazine, April, 1942.
[8] Technical Advisory Board of Anthracite Industry.
[9] *Principles of Economic Heating* by National Association of Building Owners and Managers.

We have then a problem with many ramifications. *Not all of these conservation measures can, of course, be applied to any one installation.* However, a successful use of the facilities and knowledge at hand could probably save 30 per cent of the 180,000,000 tons of heating coal and 30 per cent of the 170,000,000 barrels of heating oil. This represents one of the greatest conservation possibilities we have ever faced: the possibility of saving 54,000,000 tons of coal and 51,000,000 barrels of oil. There is little need to translate this into units of labor and transportation. Nevertheless, it represents 214,200 tank cars of oil and 1,080,000 car loads of coal. This is indeed a challenge to all who are in a position to help.

Now, what plans are on foot to obtain as much as this 30 per cent conservation as is practicable? There are many in the War Production Board who are thinking of this problem. However, one must not expect to find any set pattern or procedure for such a major task. The Conservation Division is one of the many agencies in Washington involved. We are an advisory and not an operating agency of the War Production Board. In this role, we are constantly seeking help and advice in developing recommendations to the proper WPB operating divisions. Consequently, this problem represents one of our major interests at the moment. As I see it, the problem breaks down into four parts:

1. **Increasing the efficiency of the fuel consuming unit and heating system by: *a.* Inspection and adjustment; *b.* Providing the necessary automatic controls.**
2. **Preventing heat losses by: *a.* Promoting proper home management; *b.* Weatherstripping; *c.* Storm doors, sash and entrances; *d.* Insulation.**
3. **Relieving oil consumption in critical areas by conversion to coal in: *a.* Large apartment houses; *b.* Institutions; *c.* Commercial and industrial buildings.**
4. **Reducing total fuel consumption by: *a.* Eliminating unnecessary lighting; *b.* Eliminating casual heating of non-essential buildings; *c.* Proper use of portable radiant heating units.**

A nation-wide fuel conservation program has already been started by means of radio and newspaper campaigns. We hope that it will develop along the lines indicated above with the intelligent cooperation and assistance of all interested parties.

In the case of part 1 of the program (increasing the efficiency of the heating system) it will be necessary to obtain the most wholehearted cooperation, not only from industrial technical associations such as yours, but from all interested government agencies. A decision on this part of the program will soon be made.

On part 2 (preventing heat loss), there is, of course, no limit to the amount of publicity that can be given to some of the points since they involve no critical materials or labor. Some of them require only a little common sense on the part of the person who desires to help his country, and at the same time, his pocketbook, by proper home management. We are not yet certain, by any means, that even sufficient lumber for the necessary storm windows can be allocated. Storm windows, to a considerable extent, require the same type of lumber as is used for boxing and crating, and for which there is an increasing demand. There is more lumber at the present time going into boxing and crating alone, than was used in all construction in recent years. However, there are innovations that can be used as substitutes for regular storm windows. Cellophane pasted on cardboard frames, or by some other means attached to the inside of the window frame, will accomplish the same conservation that storm windows will give. This saving reaches an unbelievably high figure and should be number one on each home owner's program for the coming winter. Although some types of insulation are critical, there will be plenty of both mineral wool and rigid insulation. The labor involved for some of this work, however, may be critical. Therefore, it is advisable to concentrate on those portions of the house which are most readily insulated, for instance, the attic floors.

Part 3 of the program (converting from oil to coal) again involves the use of highly critical materials and facilities. Conversion from oil to coal in those areas where oil shortages exist will be necessary and has already to a large extent taken

place. In fact, conversions already in place have reduced the total oil consumption at a rate of 44,000,000 barrels per year.[10]

Part 4 of the program (reducing fuel consumption) has tremendous possibilities. The elimination of unnecessary lighting, it is estimated, will save nearly 2,000,000 tons of coal, 600,000 barrels of oil and 6,000,000 cu ft of gas. This will reduce the transportation load by an estimated 630,000,000 ton miles of coal transportation and 36,000,000 ton miles of oil transportation. It would, of course, involve great inconvenience. But war is full of inconveniences and this program is seriously being considered. There are also thousands of buildings and halls which are only occasionally used, but which require a disproportionate amount of fuel to heat them up for this short time. This practice should be discarded for the duration.

Such is the program and such are the thoughts which are passing through the minds of many of us in Washington. If they sound drastic, bear in mind that we have the tremendous goal of a 30 per cent reduction of fuel before us. It is a goal worth lifting our sights for. You heating and ventilating engineers know better than any other group the difficulties we face in deciding the course to follow in working out this program. We need your help. Your suggestions will be enthusiastically welcomed. More than ever before the War Production Board looks to groups like yours for help and guidance. You have already been generous in your assistance. We urge you to even greater effort in our common cause.

The second session was called to order at 2:00 p.m. on June 7, by Pres. M. F. Blankin and he introduced Dean John A. Goff, Philadelphia, who presented his paper on the subject of final values of the interaction constant for moist air, with J. R. Andersen and S. Gratch as co-authors.

At the conclusion of the discussion, President Blankin acknowledged Dean Goff's expression of thanks for the cooperation of the Society and said that they had made a splendid contribution to the fundamental research sponsored by the A.S.H.V.E.

The meeting was then turned over to First Vice Pres. S. H. Downs, Kalamazoo, Mich., who presented J. N. Livermore, the author of the paper on the study of actual vs. predicted cooling load on an air conditioning system.

Dr. B. M. Woods, Department of Mechanical Engineering, University of California, presented the paper on spray nozzle performance in a cooling tower which was prepared by L. M. K. Boelter and S. Hori.

President Blankin resumed the Chair and expressed the thanks of the Society to Dr. Woods and the authors for their paper. The meeting adjourned at 4:15 p.m.

On Tuesday morning, June 8, President Blankin opened the third session at 9:30 a.m. and he announced the personnel of the Chapter Development Committee as follows: W. A. Russell, Kansas City, *Chairman;* Albert Buenger, Cincinnati, president of Cincinnati Chapter; and C. E. Price, Chicago, president of Illinois Chapter.

The personnel of the Resolutions Committee to report at the final session was also announced: H. H. Erickson, Philadelphia, *Chairman;* M. W. Bishop, Milwaukee, and M. B. Shea, Detroit.

Second Vice-Pres. C.-E. A. Winslow took the Chair and introduced Prof. J. R. Fellows of the University of Illinois, who presented his paper on the use of the down-draft coking method for smokeless combustion, which he and J. C. Miles prepared.

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[10] P.A.W. release 169, May 20, 1943.

The paper on a field study of comfort reactions of apartment dwellers under fuel oil rationing was presented by Sallye Hamilton, the author (complete paper published in June 1943 A.S.H.V.E JOURNAL SECTION, *Heating, Piping & Air Conditioning,* p. 311).

WHARTON CLAY, New York, N. Y.: Inquired whether further studies were contemplated. He suggested that in buildings such as the Parkchester Apartments, New York City, in which insulation was used between the plaster and masonry walls, there might be a different comfort reaction at lower air temperatures. Neither Miss Hamilton nor Chairman Winslow knew whether further studies would be made.

M. K. FAHNESTOCK, Urbana, Ill.: Stated that the paper brought out that occupants cannot be physiologically comfortable in temperatures below 73 to 74 F and that increased clothing must accompany fuel oil conservation. The effect of the unit of thermal insulation depends on its location on the body and is greatest on the extremities, particularly the feet. He questioned the desirability of introducing new units such as clo and met which might cause confusion because of the attempt to combine physical science and biological science.

CHAIRMAN WINSLOW: Stated that the units met and clo had been selected (at a conference of three of the leading laboratories working on clothing problems in the United States and Canada) for use in making clothing studies only.

H. F. RANDOLPH, Utica, N. Y.: Asked whether anything had been done to introduce shades, drapes, storm windows, etc. and to determine their effect.

E. K. CAMPBELL, Kansas City, Mo.: Stated that the real opportunity for fuel oil saving occurred in office buildings. Since the quantity of oil used for human comfort is extremely small—2 million gallons in Kansas City as compared with 140 million gallons for all purposes—a 10 per cent saving in oil used for heating would be a very small percentage of the total oil used.

CHAIRMAN WINSLOW: Called attention to other housing projects as well as Hillside Homes where a saving of fuel oil was always accompanied by an increase in consumption of gas and electricity, which might be of greater importance next year. He thought that insulation and clothing were the really effective means of meeting the situation of fuel oil reduction.

C. E. SHAFFER, Kearny, N. J.: Inquired whether the health of occupants was affected.

H. M. HART, Chicago, Ill.: Inquired why the project had not been converted to burn coal.

Miss Hamilton closed the discussion by stating that all buildings in the Hillside project had been converted to coal except one in which there was some special reason for continuing to use oil. She gave several illustrations of occupants' attempts to increase their comfort by reducing heat loss and mentioned need for educating the occupants in the matter of conserving the available heat. There were instances in which sickness seemed to be traceable to the reduced temperatures.

The third paper on the program was announced by Chairman Winslow who introduced Prof. F. B. Rowley for the presentation of his report on heat transmission through insulation as affected by orientation of wall, which he prepared with C. E. Lund as co-author.

Professor Winslow introduced P. D. Close, who presented his paper on the graphical method of calculating heat losses.

President Blankin resumed the Chair and said that he wished to make a special announcement of great importance to Society members. He stated that at the January Council Meeting it was decided to enlarge the scope of the Research activities of the Society, because it was felt that this was vital to the life and progress of the Society.

It was concluded that a qualified director of research, who could combine the duties of administrator of the Research Laboratory, liaison officer between the cooperative research organizations of the Society and be in direct charge of the development of the Research program, was needed so the new position, that of Director of Research, was created by Council. The Chairman of the Committee on Research and the Research Executive Committee, were authorized to secure a man with the proper qualifications and after diligent search the Committee and the Council feel that they were most fortunate in being able to choose Cyril Tasker, Senior Research Fellow of the Ontario Research Foundation, as the new Director of Research.

C. M. Ashley, Chairman of the Committee on Research, escorted Mr. Tasker to the rostrum. President Blankin expressed his pleasure in being able to welcome Mr. Tasker as the new Director of Research and in response Mr. Tasker said:

Mr. President and Members of the Society, I think your President has rather put it the wrong way around. The honor is entirely mine. I very deeply appreciate the trust which the Society through the Council and the Committee on Research, has seen fit to put into my hands. I can pledge you one thing, and that is my whole-hearted and undivided attention to my duty. I can pledge you a sincerity of purpose. I can pledge you that the Society's name shall be very jealously guarded in all that I do on its behalf. I ask from you your continued friendship and your whole-hearted support. This is a team job.

"We have a tremendous amount of work to do, and we must make sure that our Society's technical activities and the research work stand high, not only on the North American continent but also across the water. I am going to do my best to make certain that we do that. I hope to be able to take over my duties in September, and I hope to be able, at the 50th Annual Meeting, to report some reasonably good progress. Thank you very much indeed."

The meeting adjourned at 12:45 p.m.

Pres. M. F. Blankin called the fourth session to order at 2:15 p.m., Tuesday, June 8 and the first paper was on the subject of economic factors in converting recirculated air for ventilation and was presented by the authors H. E. Ziel and Henry Sleik (complete paper and discussion published in JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July 1943, p. 367). Mr. Sleik gave a discussion of general ventilation requirements and air treatment for air recirculation and Mr. Ziel gave the design problem involved in the practical application in a typical war plant.

PANEL DISCUSSION

President Blankin invited John Howatt, chairman of the Panel Discussion on, Will Current Ventilating System Operation Undermine Public Health and Efficiency, to present the other panel members. John Paul Jones, consulting engineer, Cleveland, L. L. Lewis, Syracuse, N. Y., and Dr. C.-E. A. Winslow, New Haven, Conn., Second Vice-President of the Society were introduced.

CHAIRMAN HOWATT: As the announced subject was selected because of a growing practice in government-financed defense projects and FHA community houses, of curtailing the amount of outdoor air brought in for ventilation purposes to as low as 5 cfm per person, discussion must center around the question of whether 5 cfm of outdoor air, per person, is adequate for ventilation in confined spaces intended for human occupancy.

The past researches and experiences of our Society seemed to show that there are minimum quantities of air supply to enclosed spaces below which it is not safe to go. The A.S.H.V.E. ventilation standards, which are the standards today, are based upon research and experience which proved at the time when they were adopted that not less than 10 cfm of outdoor air, per person, is required to maintain adequate standards of temperature, quality, air motion and distribution, and those 10 cfm are the minimum.

However, can we afford to depend too much on the past? Is our past experience to furnish the light for our path ahead? Sir Stafford Cripps, the War Cabinet member for Great Britain, stated recently that we are building up a vast store of knowledge for the purposes of war that we will use in the time of peace. It may be that the ventilation practices and changes that are now taking place in war projects will have an influence on our standards and our practices after the war. The degree to which practices in lowering the standard of minimum ventilation requirements affect health and efficiency of workmen will, of course, have great bearing on whether or not they will be continued after the war is over. What has brought about this demand today for a curtailment in the outdoor air supply? What is the reason for it? Is it entirely a question of economics? Of course we know a great many things are being done today in the interest of expediency, and properly so, because we must meet an emergency with emergency measures. But is this exactly an emergency?

No single standard will meet every need. Ventilation is more than a quantity of air. It is the maintenance of a quality that constitutes the environment. It includes temperature, humidity, air motion, freedom from odors and from bacteria, dust, and deleterious gases. The quanity to meet these requirements usually far exceeds the quantity required to maintain a supply of oxygen for metabolism purposes. So in some cases we find it is a problem entirely of engineering, rather than of physiology.

In working on designs for air raid shelters we can calculate accurately the air quality that will exist under any set of conditions, because then we are dealing with gas tight enclosures. In shops and factory buildings that are far from airtight, the uncertainties of infiltration always exist, making exact forecasts impossible. How much dependence can we place upon the unpredictable? How much should be according to rule? How much should be left to good common engineering sense?

Each member of this panel will give a short prepared statement of his views on this whole subject, following which we shall enter into a panel discussion, at the close of which the audience will be invited to participate.

I will now call for a statement from our panel member, Mr. Jones.

MR. JONES: A good many years ago I had the privilege of attending a course in heating and ventilating under Professor Carpenter, who was then the Dean of Engineering at Cornell, a very eminent authority and author of most of the early books on the subject. In one of his books he stated that good country air contained about 2 parts per million of carbon dioxide, and he advanced the theory that humans could live without ill effect in a concentration containing up to 7 or 8 parts, and from that he established what he called the standard of purity.

Now, with the knowledge that the average human exhales something like six-tenths of a cubic foot of carbon dioxide per hour, it was an easy matter to calculate the quantity of fresh air required to maintain a standard of purity, which he stated was 7 parts per million.

That was the basis on which most of the earlier school codes were established, because it happened to work out about 30 cfm of fresh air per person. In Pennsylvania the code used to be 30 cu ft. In Ohio it was six changes, which, with the limitations on cubage, amounted to about the same thing. Since that time we have definitely established that human beings can live in far greater concentrations of carbon dioxide. In any building, where there is a reasonable cubage per person and some infiltration, the supply of oxygen can be entirely neglected. There is no reason why we could not safely go to complete recirculation without any mechanically-supplied fresh air, provided only that proper temperature, humidity and control of odors are maintained.

It seems to me, therefore, that this discussion resolves itself into the question: Can you maintain proper air conditions with as little as 5 cu ft per person? I am convinced in my own mind that at least in many of the so-called defense plants that is entirely impossible. I am thinking specifically of a plant that was built a short time ago in Cleveland, which has a cubic content of about 7 million feet and a population of perhaps 12 to 15 hundred. Incidentally, it is a windowless building. We figured the zero weather heat loss at about 6 million Btu per hour, and the internal heat load, including only lights, people, and power, at something over 9 million. The plant is ventilated with an atmospheric cooling system in which they supply almost 1¼ million cubic feet of air in summer, all of which is outside air, naturally. In winter that is cut to about 700,000 cu ft, and enough outside air is taken automatically to maintain a 70 deg temperature.

It was expected that in zero weather it would require about 65 deg entering temperature (10 to 15 per cent of outside air), to hold the temperature down. This would be equivalent to 45 cu ft per person. In actual practice the plant has been somewhat overloaded with both men and machines and we have found that in 30 deg weather it takes 55 deg entering temperature, which is about 37½ per cent outside air and provides about 75 cfm per person.

I am rather opposed to codes, on general principles, and I doubt very much if it is possible to set up a standard which is applicable to all such jobs.

CHAIRMAN HOWATT: Mr. Lewis, will you give us your opinion?

MR. LEWIS: Mr. Referee, my remarks are directed toward ventilation only, rather than the total volume of air. I will deal specifically with that. In connection with it, I don't believe that we can sit here and draw any general rules of ventilation that will apply straight across the board to the question, "Will reduced outside air quantity undermine public health and efficiency?"

The reason is that in all buildings where people assemble for pleasure, commerce, or work, they foul their own air, or it is fouled for them to a greater or lesser degree; depending upon many different factors. There is no general rule of 10 cfm or any other volume which will apply to all. I have engineered plants in which 5 cfm was ample; others in which 50 cfm applied by conventional methods was entirely inadequate and still others in which the requirements of an industrial process would provide more than 250 cfm per person unless extreme measures were taken to make dampers abnormally tight.

Let's examine some of the factors which apply. There are cases in which the very minimum of expected infiltration will give super-ventilation. There are buildings the cubic contents of which provide reservoirs, replenished 24 hours per day by infiltration and of such size that they will not be overly contaminated during normal period of occupancy. There are rooms in which 10 cfm is ample in one part, and entirely inadequate in another. There are buildings, particularly industrial plants, where satisfactory results can be obtained only by collecting contamination at the source.

In view of this, it is in order briefly to review progress.

There are still in existence codes which require the introduction of 30 cfm of outside air per person, and many installations made under the dictates of these codes are not operated even in normal times.

In the progress which we have started to make we have departed to some degree from thinking of ventilation as a percentage of the total volume of air; the total volume determined in large part by factors wholly unrelated to the requirements for ventilation. Thinking along those lines, however, is still being done.

We have largely departed from the criteria of determining ventilation by air change, regardless of the relation of the total space to the number of people in it.

We have determined the minimum volume of outside air for the individual, but we have set this up as a general rule without due allowance for the reservoir in which that individual is frequently living or working.

What should we do about this? First of all, more yardsticks should be developed and improvements should be made in those now available. More cooperation and better coordination should be called for of all of the sciences which can contribute. The overdoing which leads to the disuse of equipment should be avoided, and which, if more closely sized, would not be abandoned on account of high operating cost.

The word *comfort* should be dropped from our vocabulary and one selected that adequately fits the application of our art to the conditioning of people and apply that with an intelligence equal to that which is applied to an inert substance in process in an industrial plant.

As to codes, perhaps Mr. Jones and I can get along without them but maybe we need them for you.

DR. WINSLOW: Mr. Chairman, I am sorry to put you in an embarrassing situation. I may say that Mr. Howatt pleaded with us at breakfast this morning for a row—he wanted a scrap. But if he wanted that he should not have asked three different people from different parts of the country to prepare written statements beforehand. Since he did so, I think I shall have to present mine, which is essentially identical with the first two that you have heard.

The question whether the supply of outdoor air should be reduced from 10 cfm (or from 30 cfm) to 5 seems to me quite meaningless if the proposed standard is to include such occupied spaces as FHA homes and munitions plants of various types.

The present Society standard of 10 cfm is based on calculations which refer to an air-tight space in which such an amount of air is necessary in order to dilute body odors. Even in homes its application cannot be justified without consideration of many complicating factors.

The report of the Committee on the Hygiene of Housing of the American Public Health Association, which is probably the most authoritative document in this field has this to say on the matter:

"The odors given off from the body have been proved to exert a definitely harmful influence upon appetite and therefore upon health. With persons of reasonable cleanliness the dilution of these odors will require an air change of 10 cfm per person.

"Such an air change as this, with any ordinary type of construction, will be automatically attained in cold weather by normal leakage through walls and ceilings of ordinary porosity and around normally constructed doors and windows, provided the cubic space per occupant is 400 cu ft in any occupied room, and that the normal ratio of fenestration is supplied. The necessary air change can be secured in summer by the opening of windows. Since this minimum of 400 cu ft is demanded by other fundamental needs to be discussed in later sections, no other provision for air change need ordinarily be made in the low-rent dwelling. If the other fundamental needs could be met, and if dependable artificial ventilation were provided, a lesser air space might be permissible."

In the ordinary dwelling house, therefore, the factors of leakage are with ordinary building construction adequate to take care of a space which is not crowded. In an ordinary schoolroom window gravity ventilation supplying about 10 cfm of air has proved the most satisfactory method of handling the situation; but the supply of air must be accomplished by the use of properly designed heating appliances and deflectors for the admission of outside air so as to avoid unpleasant drafts. In a

large auditorium with 50 or more persons nothing short of mechanical ventilation supplying 20 or 30 cfm of air will suffice, since the people in the center of the room are too far away from the windows to get the benefit of a gravity supply.

To apply any such standard to a factory workroom with 6000 cu ft per person and probably with open doors and skylights would obviously be fantastic so far as the problems arising from human occupancy are concerned. Clearly in such a space as this a supply of 5 cfm of outdoor air per person would constitute an altogether negligible factor from a quantitative standpoint, and would be nothing more than a meaningless gesture.

On the other hand if the industrial processes in question produce a large amount of heat or generate noxious fumes and dusts, it may be necessary to provide artificial ventilation far in excess of 5, 10, or 30 cfm.

It is easy, of course, to attempt to settle such matters as this by arbitrary rule-of-thumb methods, but such a procedure does not in any way correspond with the realities of the situation.

CHAIRMAN HOWATT: Mr. Jones, is your fear of establishing codes merely that we may hang on to them when we should let go and let go when we should hang on? Are you afraid of them because of the time element?

MR. JONES: When you establish a minimum code it is pretty likely to become a maximum. If we set an arbitrary figure of so many cubic feet of air per person, in many cases it may be more than is necessary, and in other cases insufficient. I think it would be far better if we could establish a code on the basis of quality. In other words, we might say that it would be necessary to provide a sufficient amount of air so that the temperature would be between certain limits and the purity would be within a certain standard. The difficulty, of course, is that that sort of yardstick is too hard to apply in the field. It is too difficult to make work.

CHAIRMAN HOWATT: Is that your only objection to them?

MR. JONES: Yes. I have no other objection.

CHAIRMAN HOWATT: Don't the benefits outweigh those slight objections of yours? Weren't the codes primarily fostered with the idea of leveling out competition to some extent and getting rid of the chiselers in the industry?

MR. JONES: Codes for that purpose have been set up by the industries themselves but I think primarily a legal code is set up for the protection and health of the people. Certainly that is the purpose of plumbing codes, for instance, and I don't know why it should not be the same purpose in ventilation.

MR. LEWIS: I have been thinking of certain new devices that are growing up to be tacked onto our air conditioning system. We get a light that kills bacteria and a device for collecting the products of flatulence and then comes the improvement of building construction, glass brick, and things of that sort to make buildings tighter. After a while, we may kill all the benign bacteria along with the malignant.

We should look forward to the time when infiltration, for which we should be more grateful than we are at times, is all cut off. Then we will need codes, but our codes should be realistic with respect to the ultimate results that we want to obtain.

A code that is based purely on the amount of outside air that is taken in without regard to where that air comes from, is really silly. There might be next door to your outside air intake some of the foulest contamination, or there might be products given off within the rooms which could not be diluted with ordinary ventilation, in any volume.

I think while we are perhaps fully justified in doing a little smiling at codes, it is nevertheless quite essential that we keep them around and nurse them along until we can make them realistic.

CHAIRMAN HOWATT: What would the Health Department do without a code?

MR. LEWIS: What would we do without a Health Department?

CHAIRMAN HOWATT: Very poorly.

MR. JONES: I think that last remark more to the point.

DR. WINSLOW: I should like to reinforce what has been said by an analogy. I have no objection to codes, but I think they should be sensible. Suppose in a problem of heating you attempt to operate on the basis of supplying so many Btu per person per hour, irrespective of what happens outside or what happens inside or the conductance of the building? You don't do that. That is no sillier than it is to require 50 or 30 or 10 cfm without taking into account any of the conditions that are operative. It is an attempt by a shortcut to dodge a fundamental engineering problem.

Now, you take a contract to provide a certain temperature under certain external weather conditions in a particular building, and you work that out, and you determine how many Btu are needed. I think the same principle is the only one that will solve this question of air change.

CHAIRMAN HOWATT: Dr. Winslow, when a contractor is dealing with the unpredictable, which in this case is an uncontrollable air supply, just how would he arrive at a bid to guarantee end performance?

DR. WINSLOW: Just as he does in leakage. You can predict leakage. You can specify certain conditions of occupancy, obviously, just as you do in your heating.

CHAIRMAN HOWATT: The heating load, of course, is determined by factors from our Guide, based on a certain anticipated amount of infiltration. You set up for a peak load condition then, a condition under the worst that will arise.

DR. WINSLOW: Assuming that temperature is taken care of, but with no other problem except avoiding the accumulation of objectionable odors from the human body, you need 10 cfm per person of air change, but you ought to allow for presumable leakage and, what is much more important, on the other side you have got to allow for other sources of odor, for smells from cooking and for excess heat produced and for fumes and so forth. The engineer cannot dodge his responsibility of studying the particular building that he is working on and deciding on the basis of his best judgment what is needed.

CHAIRMAN HOWATT: When an engineer cannot do the engineering himself he specifies that the contractor or the manufacturer shall maintain certain results. That is an old game. Those are *murder* clauses still found in many specifications. Well, are you in favor of that kind of *out* for engineers who are afraid to take the responsibility for the solution of their problems?

DR. WINSLOW: No. I think the engineer has got to exercise professional judgment—not the contractor; but I don't think anyone but the engineer can deal with such situations as Mr. Jones has mentioned. It has got to come back to the wisdom and experience of the designer.

CHAIRMAN HOWATT: Well, then the designer has to set up the air quantities, not the results.

DR. WINSLOW: Well, he sets up the air quantities on the bases of the results to be obtained, just as you set up your design for heating on the basis that you want to maintain, say, a 70-deg temperature.

CHAIRMAN HOWATT: Speaking not as an officer of the Society, Doctor, but as a well versed member, the Society has several code committees working on codes and on adjustments of codes at this time. Are you in favor of the elimination of all of our codes and the discharge of all of these committees?

DR. WINSLOW: As far as I know, most of the codes operate on the basis I have been speaking about. I have no attack to make on codes. Most of the codes specify the type of construction which will attain certain results.

CHAIRMAN HOWATT: They are dealing with unpredictables, are they not? In my opinion, the real objection to a code is that the thing lasts too long. It generally outlives its usefulness long before it is thrown out and, because it outlives its usefulness, to that extent it does hinder progress. That is my main objection to codes. They stay and they stay in our thinking and our practice long after they have ceased to be useful. Look how long that 30 cfm per pupil stayed in schoolhouses.

DR. WINSLOW: It is still there.

MR. LEWIS: After a while the code just stops you from thinking.

CHAIRMAN HOWATT: Now, Dr. Winslow, you are known as an authority on physiological reactions of the human system to its environment, a health officer who can advise the Society and guide it in the ways of health. There are three phases of ventilation problems: one is health, another is efficiency of the workman, and the other is comfort. In total war we know that it is essential that a worker producing food or munitions of war must receive the same care that a soldier does on the battlefield, if the wastage of our manpower is to be prevented. How important the health of the people is to the welfare of a nation is indicated by the principles laid down in the Four Freedoms, two of which, freedom from fear and freedom from want, are definitely important health factors. Of course, health cannot be measured like the pressure on a steam gage, but there are certain indices that can be used to determine health, determine disease, and there are tables of mortality. They do not tell us exactly what caused the sickness or the death but they do give us material on which we can think about those problems. Dr. Winslow, will you tell us what you think about the health phase of ventilation?

DR. WINSLOW: Well, I think perhaps the best answer to that is to point to the winter prevalence of respiratory diseases. It is the most interesting and important fact in epidemiology. Why do we have colds and pneumonia and influenza in winter and not in summer? That is a problem of air conditioning. There is no other reasonable explanation of it. I think that is an illustration of the very important effect on health of too great shocks from the temperature angle. You get just the same thing in certain industries. I hope there are no reporters present who can misconstrue this, but you get the same excess death rate from pneumonia among workers in blast furnaces (in places other than Pittsburgh, let us say) who are supposed to be exposed to sharp contrasts between heat and cold.

You mentioned efficiency. Some of our best knowledge of the relation of temperature to efficiency grew out of the English studies during the last war, when they showed beyond question the direct effect on production, the very important effect on production of the maintenance of proper temperature in the workrooms.

CHAIRMAN HOWATT: Now, you see, Doctor, it is easy for us to say *ventilation* and *proper air environment* are so important to health. Isn't health a very difficult thing to measure, does it not take a long time before any influence is noticed from an air environment on the health of the people? Have we been emphasizing too much the importance of ventilation on public health?

DR. WINSLOW: Well, I think not too long to be measurable in a great many ways.

CHAIRMAN HOWATT: The other factor that is involved in this question is the factor of comfort, the air environment with relation to comfort. Mr. Lewis said he wished the word *comfort* had been left out of our vocabulary. Don't you feel comfortable today? Why don't you like the word *comfort?*

MR. LEWIS: Well, I don't like it because it casts a fog around, or a halo over the true objective. I believe that one of the great misfortunes of our profession and industry is the almost universal use of the word *comfort* to describe either the objective or the results of air conditioning or ventilating or heating. This misfortune applies not only to those who make their living out of the industry, but also to those who should benefit from the use of our services and products.

No one can justly single out anyone to be blamed for this because it just grew and no one could foresee the outcome. Our Society created a comfort zone on a chart. Hundreds of individuals have applied the word *comfort* thousands of times. I know only one who has tried to lead us away from its use—Charlie Leopold.

The upshot of it all is that we have held the spotlight on the erroneous conception that the prime and perhaps the only function—particularly of air conditioning—is to provide rocking-chair comfort.

This false concept has too completely obscured the real objective of raising the

quantity and the quality of the production of human beings. It has led to a misguided condemnation by high authorities which have added many difficulties to getting done those things which are essential to the quantity and quality of essential war production.

Let's bound the field with a few typical examples.

Many vital parts of bombsights, aircraft instruments, radio, radar and the like are made of non-ferrous metals. On these precision parts, the finger print deposits of skin excretion frequently become the seed from which corrosion may grow when the part is exposed to an environment favorable thereto—a growth which, without any warning, impairs or destroys the usefulness of the part.

The quantity of skin excretion—not visible enough to be called perspiration—is increased or diminished by environmental air conditions. The objective in conditioning these production plants is simple, sound and obvious and is not to be confused with or obstructed by any thought of rocking-chair comfort.

In conditioning a bank, the real objective is to reduce the cost of doing a banking business. It is accomplished by providing an environment which will reduce the number of errors, increase the production of the individual and lessen turnover through improved customer employee relationships.

The bank can measure production in terms of items per employee and can get its answer in a black ink balance sheet, expressed in dollars and without regard to units of rocking-chair comfort.

The objective in reducing the gaseous and particular contamination in a welding shop is to increase production and improve the quality of the product—and, of course, health.

Although it is definitely out for the duration—applying complete air conditioning to the home comes close to the rocking chair but the analytical thinker will look beyond it to two ultimate objectives, hastening recuperation from the day's fatigue and rebuilding reserve capacity for the next day's production.

Dozens of examples can be cited but these four cross the field from the four points of the compass.

No, our ultimate objective is not comfort. It is to give man or woman a fair chance to apply the fundamental of creating wealth by producing up to the limit of his latent capacity.

In working toward this objective, there are two channels of design. One is just to condition the space—the other is to condition the human being who works in that space. The first is shortsighted, for provided we give the individual human being good air to breathe—what does it matter if environment 10 ft away is not good—provided we sweep away his animal heat at a commensurate rate—and provided we protect him from the heat or from the cold of his environment.

We could do better if we would alter our fundamental thinking, find a new and appropriate word to define our objective and adjust our practice accordingly.

CHAIRMAN HOWATT: These are certainly new ideas. The panel at least is not in a groove throughout on codes or on the word comfort. As far as I am concerned, it seems to me if I am too comfortable I get lazy. The third step in the need for a proper air environment, Mr. Jones, I am going to ask you to discuss a bit, and that is what effect it has on the output of workmen, the so-called efficiency of the plant or the man, and what relation does it have to some other factors that you may have in mind.

MR. JONES: I have a pretty strong conviction in my own mind that health and this word that Lewis does not like, *comfort,* and efficiency, are all pretty much wrapped up with each other. That is, I have a notion that an atmosphere which is comfortable is also healthful, and probably promotes efficiency. Although unfortunately, there is to my knowledge not much statistical data on the subject of efficiency of workers, I still believe it is a real thing.

I know of one example—an inspection department of a plant which was made up largely of girls—which was moved from a very poorly heated and usually over-

heated, unventilated, inadequately lighted room into a brand new, air conditioned space, with an illumination of about 50 foot-candles on the working table.

Immediately, the number of pieces of material that passed over those desks increased about 60 per cent. It is quite obvious that some of the improvement could be charged to air conditioning, but how much is chargeable to air conditioning, how much to better light and how much to the simple fact that they might have had a lot of work to do in a hurry, is hard to say.

Incidentally, that question of efficiency is one which is important to those of us who make a living, or at least a part of a living, by designing or selling or installing air conditioning equipment, because it creates a very handsome yardstick to measure the value of a set of equipment to an industrial plant. I wish that we had more information on the subject, because it would be a fine sales talk.

CHAIRMAN HOWATT: Do you want to challenge him in any way, Dr. Winslow?

DR. WINSLOW: No. I think I am entirely in agreement with the concordance of comfort, health and efficiency. Take one illustration. We used to talk a good deal about 68 deg as the ideal temperature, without very much basis for it, because there had not been very much work done. You will remember the Society carried on some very extensive studies in factories and found that in some of them the workers lightly clothed and doing light work preferred a temperature of about 72 deg. Very careful studies in the Laboratory have shown that that temperature of 72 deg for a person lightly clothed and doing light work, or no work, was just the temperature at which the body was in equilibrium, without having to produce extra heat to give to the environment and without having to produce sweat for an extra cooling effect. I feel very sure that these problems of health and comfort and efficiency are very closely related, and on the efficiency side, as I say, there hasn't been much done in this country, but Vernon in England accumulated a lot of data, very good data, in which nothing was varied but the temperature of the factory air and in which he got very important effects on production—very material increases in production by preventing in that case overheating.

CHAIRMAN HOWATT: The panel discussion is now open for general comments.

C. S. LEOPOLD, Philadelphia, Pa.: I have two questions. First, on the question of code. I believe, Mr. Moderator, that the thing to which most of us object is being told how to do it instead of being told of the result we are to obtain. Unfortunately, I think that we will have to admit that, at this moment, we are not too sure of the result we are trying to obtain and therefore temporarily we have certain codes which are based on method. I think that is the basis of our objection to codes—we don't want to be told how.

The second point: In the preceding discussion I have noticed that the introduction of fresh air has been made synonymous with the reduction of odor. Actually, the factor of relative humidity is almost of equal importance—it certainly is of comparable magnitude. We are all familiar with the fact that if you permit relative humidity to get appreciably above 50 per cent, you have a musty odor regardless of fresh air quantity used. I wonder if Mr. Oscar Levant Lewis would like to speak on this point?

MR. LEWIS: The particular experience was with a building in San Antonio, Texas, a 20-story office building. It had quite an outside air intake but was designed for a graduated amount of outside air. After a couple of years of very successful operation we began to get complaints about odors, odors of linoleum, odors of paint and books and various other things, coming from all over the building. We sent three different men down to diagnose the trouble and prescribe the remedy. They all came back licked. Finally we tried another one. This fellow had primed himself pretty well. He knew what dewpoint and what temperature and what relative humidity was supposed to be carried in the building. He discovered that during the intermediate season, in an attempt to get rid of odors, the outside air damper control had been disconnected and the dampers blocked wide open, so instead of getting 5 or 10 cfm per person in that building we were getting 40 or 50, and

still we had the odors around. The remedy was to put the dampers back under control, to reduce the outside air quantity down to a relatively insignificant amount in relation to what had been taken in and to get the dew point and the relative humidity down.

W. L. FLEISHER, New York, N. Y.: I don't agree with your panel. In the first place, I think that a code is more than essential. Make it as minimum as you like, but Mr. Lewis is talking here from the light of his 50 years of work in this business and consequently he knows a great deal more than does the average person who has to qualify in various parts of this country and who needs a code.

Now, because of the fact that 30 cu ft of air per person had been used for so many years as the amount of air required for school ventilation and because it was so criticized in the light of our later knowledge, the State of New York voided this law and passed a new law. I was called in by some of the interested parties to fight the proposed law, which simply specified that the Director of Public Education of the State of New York should determine what was required for proper ventilation of schools. Certainly 10 cu ft of air, which we tried to have introduced into the code as a minimum, would have been infinitely better than no stated amount, and the amount to be used could have been raised above 10 cu ft but could not have been reduced below that.

After all, the reduction of ⅔ in the amount of fresh air required was going a long way towards correcting an obvious fault. The 10 cu ft had been determined by a group of engineers as being a good minimum and about what was essential. Remember that when you are dealing with the public you are dealing not with engineers, but with people who think they have a good bit of knowledge about a subject and don't require the services of an engineer. It is well to hold those people to a minimum as long as the minimum within our knowledge is not excessive. I am all in favor of codes as long as they are not extravagant in their demands.

Mr. Lewis seemingly has completely gotten off the subject of this forum. As far as the comfort zone or the comfort requirements as developed by the Society are concerned, in my opinion they have had more to do with maintaining the prestige of air conditioning and increasing your business than any other one thing in the whole range of engineering. These developments of the Society have made the public realize that there are certain conditions which are comfortable, and these conditions have been demonstrated by the Society over and over again to the point where the public believes in them. The work of the Society has also definitely indicated that beyond a certain point no one is comfortable and consequently a code within the range indicated is a safe and constructive code or standard to apply and to publish.

THOMAS CHESTER, Detroit, Mich.: I do not intend to attack the entire panel because collectively they seem fairly formidable and all of the same mind. I just propose to pick on Mr. Lewis. I am surprised to find that he confuses comfort with ease. When a man is sitting relaxed in an armchair, his state is one of ease and not necessarily of comfort; as it may be too hot or too cold for comfort. Human comfort is the fundamental objective of this Society, the aim being to provide a condition as regards air temperature, air movement, vapor pressure and mean radiant temperature which closely approaches the ideal. I see no reason for scrapping the word *comfort* as it is certaintly expressive and I suspect that Mr. Lewis is just amusing himself in suggesting it. In the sense that we use the word comfort, it pertains to disposal of the heat which must be lost by the body, without any consciousness of heat or cold or of perspiration; in other words with no discomfort. With suitable atmospheric conditions energy is increased. An individual can be numbed with cold or weakened by heat. In hot countries people refrain from working in the hottest part of the day; they take siestas.

With Dr. Winslow present I do not feel like saying much about physiological matters but it is now common knowledge that in too hot an environment too much blood goes into the sublayers of the skin in an effort to lose heat and not enough to the vital organs.

I favor keeping the word comfort in our vocabulary.

L. T. AVERY, Cleveland, Ohio: If you are going to talk about the quantity of air as being the ventilating system operation, that is, 10 cu ft, or 5, or 30, I don't think it has anything to do with operation. If there is one thing that causes odors, that would make a place distasteful from a working comfort standpoint, it is the stink—and stink has more to do with cleanliness of the apparatus than it does with the quantity of outside air. Nothing in our standards provides any means at all for regulating how often this equipment should be cleaned, or what is clean air. So if we are going to standardize, and say 5 or 10 or 30 cu ft and be silent on the rest of it, we have missed the boat. I for one don't think we are undermining public health and efficiency and it will be too bad to let either subject drop without getting that pretty clear that we are not undermining anything; we are simply neglecting ordinary housekeeping.

CHAIRMAN HOWATT: Are there any other questions from the audience?

H. M. HART, Chicago, Ill.: I cannot agree that we should abandon codes. We worked a long time in Chicago trying to write a city code for ventilation, and I was on the committee. We tried to incorporate in the city code the American Society of Heating and Ventilating Engineers' Code which specified the results to be obtained but the authorities and the Health Department said that they could not administer it. They were so violently opposed to it that the only thing they would consent to do would be to try to calculate the volume of air that would be necessary to produce the desired results for different types of buildings, and put that in as the code required. The committee of engineers who helped to formulate the code did not agree with this procedure, but from an administrative standpoint we had to back down. I must admit that I don't know how they could administer it. The administrator would certainly have gotten into a lot of trouble if the code simply provided for certain results to be obtained and after the building was erected the system failed; it would mean that he would have to condemn the building and have it torn down and built over again. I don't think he would have lasted very long under those conditions. That was our difficulty. And so we concluded that the only way out was to try to calculate the minimum air quantity that would be required to maintain the conditions that they hoped to obtain.

MR. LEWIS: In regard to this business of comfort—an ideal working environment is not one in which you are conscious of being comfortable. It is one, to borrow from Charlie Leopold, in which you are wholly unconscious of temperature, humidity, air motion, or anything else. You are not annoyed by anything plus or minus.

As to codes, I think you might as well leave out of the record everything that has been said and print Charlie Leopold's remarks about codes in big capital letters; that they should not tell us how to do something, but should tell us what upper limits must not be exceeded. The how-to-do-it should not be frozen into a code.

DR. WINSLOW: Before closing I want to enter a protest against the unfair tactics of the Chairman. He selected the title of this symposium and then he wrote to each one of us and asked us to talk about 5 cfm, which, quite justly, had nothing to do with the title and then, when we presented ourselves here with a touching and beautiful magnanimity he misrepresented us, tried to make it appear that we wanted to be uncomfortable, that we were opposed to all codes, and that is manifestly unfair. Then the final error he made was in not having Mr. Leopold on the panel, because I agree with Mr. Lewis that Mr. Leopold said the one really vital thing, that codes should deal with the results to be produced, and not with the methods.

CHAIRMAN HOWATT: Just to correct the worthy gentleman on my left, the Chairman or the Moderator had nothing to do with the subject at hand. It was selected by the Meetings Committee of the Pittsburgh Chapter. Neither had I anything to do with the panel selection, and if I had, I would have chosen the same panel.

It is evident to all of you, after listening to this panel discussion, that there can be no such thing as a hard-and-fast rule about how much air shall be recirculated

per person, or how much of that air shall come from out of doors, in order to provide what is now called an ideal inside climate; 5, 10, 20, 50 cfm, any one of them may be wrong at some time. The quantity required depends on so many factors that every building should be considered as a problem by itself, studied with relation to the building construction, the kind and the type of work that will be carried on in the space. It is evident that the one-third cubic feet of air needed per minute per person to provide the oxygen required for breathing is indeed a negligible factor and need not be considered in any of these engineering problems. We use the air merely as a distribution medium to obtain results, and the amount of air that is going to be required or circulated will depend entirely upon how fast the desired conditions are broken down within the building. The decisions finally arrived at should follow an intelligent study by competent, trained, experienced engineers. So to ask if 5 cfm is enough is like asking a man if he would stop beating his wife. The question cannot be answered by yes or no.

At the conclusion of the panel discussion President Blankin expressed the appreciation of the Society to members of the Panel for a most interesting and instructive session.

The chairman of the Resolutions Committee, H. H. Erickson, Philadelphia, then gave his report as follows:

Report of Committee on Resolutions

Whereas, the Semi-Annual Meeting 1943 of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been an outstanding event in the City of Pittsburgh, June 6 to 8, 1943.

Therefore, Be it Resolved, That an expression of thanks and appreciation be adopted by this meeting and be spread upon the minutes of the Society and copies thereof be transmitted to each of the persons and agencies who have contributed to making this meeting so enjoyable for the members of the Society who attended:

To G. G. Waters, President of the Pittsburgh Chapter and the Chapter Members for the capable manner in which they fulfilled their positions as hosts,

To Chairman Ralph B. Stanger and the Committee on Arrangements whose careful planning has been shown in every detail of this meeting,

T. Mrs. T. F. Rockwell and her Committee who under trying war conditions were able to do such splendid work in entertaining the ladies,

To Frank L. Duggan, President of the Pittsburgh Chamber of Commerce for his address of welcome to Pittsburgh,

To the Authors and Panel Speakers at technical sessions for their instructive papers and able presentations,

To the newspapers and trade publications whose columns have given advance notices and daily coverage to the Semi-Annual Meeting,

To Howard Coonley, Director, Conservation Division, War Production Board, for his interesting analysis of the Fuel Conservation problem,

To the Management and all employees of the William Penn Hotel who contributed much to the success of the meeting and comfort of the members under the difficult war circumstances,

To Fabian C. McIntosh, Chairman, and the Hospitality Committee for their splendid contribution to the success and enjoyment of this meeting,

To the Pittsburgh Convention and Tourist Bureau for their cooperation and assistance,

To the nine Past Presidents for their continued interest and attendance at this meeting,

To Dr. Allen A. Stockdale for his inspiring address,

To the members of the Society who are in the armed forces and others who are giving their services in many ways, for the successful prosecution of the war.

Finally, to every member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS who has attended this meeting for his loyalty and support.

Respectfully submitted,
H. H. Erickson, *Chairman*
M. W. Bishop
M. B. Shea

On motion of Mr. Erickson, seconded by L. T. Avery the report of the Committee was unanimously adopted.

As there was no unfinished business nor any new business the meeting was adjourned at 4:15 p.m.

ENTERTAINMENT

For those who arrived in Pittsburgh on Sunday, June 6, the Hospitality Committee of Pittsburgh Chapter greeted the members and ladies for their entertainment. The ladies guided by the Ladies Committee and hotel staff enjoyed a tour of inspection of the William Penn Hotel, at 4:00 p.m. They viewed the kitchen while a meal was in preparation and saw the refrigerators, the wine cellars, linen rooms and many other intertesting sights. At 6:00 p.m. there was a reception for visiting members and ladies in the Cardinal room of the William Penn, which provided an enjoyable social hour and a buffet supper.

The Ladies Committee in charge of Mrs. Theo. F. Roçkwell, welcomed the visiting ladies and entertained them at tea at 3:00 p.m. on Monday afternoon.

After the all-day technical sessions and various committee meetings, the entertainment committee with H. E. Park as chairman, arranged an informal get-together at the Nixon Restaurant, where the members enjoyed a mid-night supper and floor show.

For the concluding event, as a fitting climax to the very interesting meeting, an informal dinner was held in the Urban room of the William Penn, and E. C. Smyers was chairman of the Committee in charge. During the dinner an orchestra played musical selections and the Television Kids from Station KDKA gave a demonstration of modern and classical music. At the conclusion of dinner Toastmaster F. C. McIntosh introduced the distinguished guests and the nine Past Presidents of the Society who were in attendance. He then called upon Pres. M. F. Blankin to present a resolution to John James, former technical secretary of the Society.

The Past President's Memory Book was presented to Prof. E. O. Eastwood of Seattle by President Blankin. In a brief speech which reviewed the professional accomplishments and services rendered to the Society by Professor Eastwood as President, F. E. Giesecke, Past President, presented him with the Past President's Emblem of the Society.

The speaker of the evening, Dr. Allen A. Stockdale, New York, was introduced by Mr. McIntosh, and speaking on the subject "Democracy Can Do It," Dr. Stockdale gave an inspiring message which was greeted with frequent applause by the audience. He described the tremendous output of industry, the colossal tonnage of shipping, guns, tanks, ammunition, airplanes, trucks, and other implements of war. He contrasted the fundamental beliefs of our democracy with those of the peoples in the dictator dominated countries, and

brought out the fact that we had freedom of action, speech, and many other privileges, that our army had an unselfish objective and was not actuated by the ambitions of a conqueror. It was a stirring address that gave the audience many facts to demonstrate that only in a Democracy could the cooperation and the will to win, wield the people into a unit that would overcome all obstacles.

SEMI-ANNUAL MEETING PROGRAM

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

WILLIAM PENN HOTEL, PITTSBURGH, PA.

June 6-7-8, 1943

Sunday—June 6

10:00 A.M.	Research Executive Committee Meeting (*Parlor C*)
1:00 P.M.	REGISTRATION (*Silver Room*)
1:30 P.M.	Council Meeting (*Forum Room*)
2:00 P.M.	Committee on Heat Transfer in Finned Tubes (*Parlor B*)
4:00 P.M.	Ladies Tour of Inspection
6:00 P.M.	Reception of Visiting Members and Ladies (*Cardinal Room*)
7:30 P.M.	Committee on Research (*Parlors E and F*)

Monday—June 7

9:00 A.M.	REGISTRATION (*Silver Room*)
9:30 A.M.	TECHNICAL SESSION (*Urban Room*)
9:35 A.M.	Welcome to Pittsburgh—Frank L. Duggan, President, Pittsburgh Chamber of Commerce
9:45 A.M.	Reports of Officers and Committees
10:10 A.M.	Amendments to By-Laws
10:30 A.M.	Performance of a Residential Panel Heating System, by H. F. Randolph and J. B. Wallace
10:50 A.M.	*Discussion*
11:45 A.M.	Adjournment
12:15 P.M.	Welcome Luncheon—*Toastmaster,* G. G. Waters, President Pittsburgh Chapter; *Speaker,* Howard Coonley, Director, Conservation Div., War Production Board; *Subject,* Fuel Conservation
2:00 P.M.	TECHNICAL SESSION (*Urban Room*)
2:10 P.M.	Final Values of the Interaction Constant for Moist Air, by John A. Goff, J. R. Andersen and S. Gratch
2:30 P.M.	*Discussion*
3:00 P.M.	Study of Actual *vs.* Predicted Cooling Load on an Air Conditioning System, by J. N. Livermore
3:20 P.M.	*Discussion*
4:00 P.M.	Spray Nozzle Performance in a Cooling Tower, by L. M. K. Boelter and S. Hori
4:20 P.M.	*Discussion*
4:45 P.M.	Adjournment
3:00 P.M.	Ladies Tea (*Suite 468-70*)
5:00 P.M.	War Service Committee (*Parlor D*)
5:00 P.M.	Chapter Delegates Conference (*Parlors E and F*)

7:30 P.M. Committee on Heavy Duty Furnaces (*Parlor G*)
8:00 P.M. Committee on Psychrometry (*Parlor D*)
8:00 P.M. Guide Publication Committee Meeting (*Parlor B*)
8:00 P.M. Nominating Committee Meeting (*Adonis Room*)
9:00 P.M. Informal Get-Together—A Night at The Nixon

Tuesday—June 8

9:00 A.M. TECHNICAL SESSION (*Urban Room*)
9:05 A.M. Use of the Down-Draft Coking Method for Smokeless Combustion, by J. R. Fellows and J. C. Miles
9:25 A.M. *Discussion*
10:00 A.M. Field Study of Comfort Reactions of Apartment Dwellers Under Fuel Oil Rationing, by Sallye Hamilton
10:15 A.M. *Discussion*
10:40 A.M. Heat Transmission Through Insulation as Affected by Orientation of Wall, by F. B. Rowley and C. E. Lund
11:00 A.M. *Discussion*
11:00 A.M. Ladies Brunch-Bridge (*Adonis Room*)
11:30 A.M. Graphical Method of Calculating Heat Losses, by P. D. Close
11:45 A.M. *Discussion*
12:15 P.M. Adjournment
1:30 P.M. Meeting of Technical Advisory Committee on Radiation and Comfort (*Parlor G*)
2:00 P.M. TECHNICAL SESSION (*Urban Room*)
2:05 P.M. The Economic Factors in Converting Recirculated Air for Ventilation, by H. E. Ziel and Henry Sleik
2:25 P.M. *Discussion*
3:00 P.M. Panel Discussion — John Howatt, *Chairman; Subject* — Will Current Ventilating System Operation Undermine Public Health and Efficiency: Prof. C.-E. A. Winslow, New Haven; John Paul Jones, Cleveland; L. L. Lewis, Syracuse
4:45 P.M. Adjournment
7:00 P.M. Dinner (*Urban Room*)
Toastmaster, F. C. McIntosh; *Speaker,* Dr. Allen A. Stockdale; *Subject*—Democracy Can Do It
Presentations: Past President's Emblem and Memory Book to Prof. E. O. Eastwood

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1236

No. 1236

PERFORMANCE OF A RESIDENTIAL PANEL HEATING SYSTEM

By H. F. Randolph * and J. B. Wallace,† Utica, N. Y.

THE objects of this investigation were to compare calculated requirements of a panel heating system using warm air as the heating medium and ceiling panels as radiators, with actual operating requirements and to observe the environment produced by such a system under both constant and intermittent operation.

Equipment and Limitation of Use

In March 1941 the erection of a typical two-story and basement eight-room frame house (Figs. 1 and 2) was completed in Utica, N. Y. The heat loss at an 80 F temperature differential was calculated as 44,468 Btu, exclusive of a 12 ft 6 in. x 23 ft 6 in. basement game room and the garage, and 54,277 Btu including the game room. The total floor area of the heated space, including the game room, was 1735 sq ft and the volume 13,052 cu ft and 1442 sq ft and 10,996 cu ft excluding the game room. The exterior wall construction consisted of wood siding on 25/32 in. fiber sheathing board, 2 in. x 4 in. studs, rock lath and plaster. Between the studs, with air space on either side, were placed 2 in. thick bats of balsa wool within vapor seals of asphalt impregnated paper. Between the second floor ceiling joists were placed 4 in. thick bats of rock wool with vapor barrier on the under surface.

All windows and doors were weather stripped. Storm doors and windows were used except on the maid's room and lavatory windows and the front door.

The maid's room was constructed under a sun deck and over a 12 in. excavation below the floor, the area way leading from the excavation to the main part of the basement being closed during all tests. The floor of the maid's room consisted of joists, sub-floor, ½ in. plywood and linoleum with no insulation. The four walls, floor and ceiling of the bath room, and floor and two walls of the kitchen were covered with linoleum.

The house, with no other structure or trees closer than 200 ft and unfurnished and unoccupied, was affected to a greater degree by solar radiation than if protected externally and having the windows fitted with shades and curtains or draperies. During one period of observation, on a clear bright day, the thermostat became satisfied at 6:48 a.m. when the outdoor temperature was 15 F and did not again call for heat until 2:49 p.m. when the outdoor temperature was 30 F. During that period of seven hours, effect of solar radiation through windows of the dining and living rooms satisfied the thermostat while

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† Research Engineer, International Heater Co.

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the temperature at the 30 in. level in the bath room, which was unaffected by direct sunshine dropped from 70 F to 65 F.

Because of such solar effect all tests reported in this paper, unless otherwise noted, were made at night starting at least three hours after sun-down, without lights or other source of heat above the basement than from the heating system.

After the first few observations the game room was omitted from test readings as, while in severe weather its temperature would reach equilibrium with the balance of the house, in mild weather it would be underheated. This apparently was due to the ground temperature adjacent to the three exposed

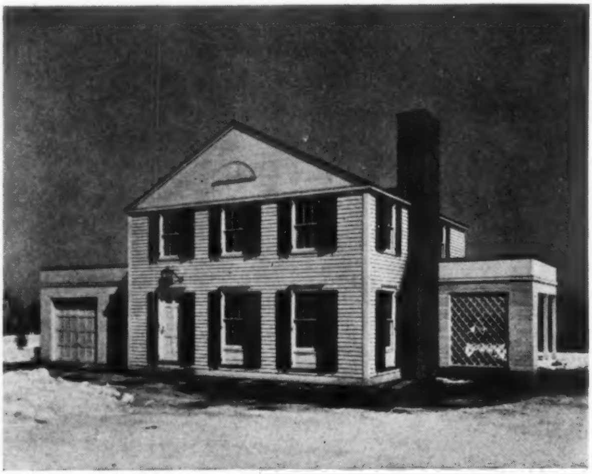


FIG. 1. HOUSE IN UTICA, N. Y., WHERE CEILING PANEL HEATING STUDIES WERE CONDUCTED

walls of the game room being more constant than the air temperature above ground, resulting in a proportionately higher heat loss from the game room as the weather moderated.

Although temperatures in the locality of this house often drop to −25 F no weather colder than 3 F was experienced during early 1941 or during the 1941-42 heating season when the tests herein reported were made.

HEATING SYSTEM AND METHOD OF CONTROL

The heating panels were constructed by stripping, with asbestos tape, the lower edges of ceiling joists and attaching thereto sheets of 30 gage galvanized iron which were also formed down the four walls of each room a distance of 2½ in. Under these sheets and to the joists were attached sheet metal U clips through which were passed 2½ in. sheet metal strips to act as guides for the air. Through perforations in the clips and strips were passed wires by

means of which metal lath was suspended. Conventional plaster was applied. Thus a 2½ in. air space above the plaster ceiling was provided.

Introduction of the air to each panel (Fig. 2) was made adjacent to an exposed wall so that the warmest part of each panel was directly over the area of greatest heat loss.

Heat was supplied by a gas fired hot water boiler connected on the supply side through a flow control valve to a copper heat exchanger and back to the boiler through a circulator (Fig. 3). The exchanger and a 14 in. squirrel

Fig. 2. Floor Plan Showing Heating Panel Arrangement

cage blower were installed in a galvanized iron duct system with one supply and one return riser to each ceiling panel so that the air was circulated in a closed system over the exchanger through the ducts to the panels and returned, with none of the air being projected into the living quarters of the house. The boiler-exchanger combination was used, rather than a direct fired unit, merely to provide a flexible means of controlling air temperature for test purposes.

From the basement trunk lines 10 x 3¼, 12 x 3¼ and 14 x 3¼ riser stacks were run to and from the various panels. One supply riser and one return riser were run to each room except the bath and second floor hall in which two rooms were heated from one set of risers. Balancing dampers were installed in the basement piping, and control dampers in the return riser from each bedroom to provide means within the room for shutting off the source of heat.

A conventional heat anticipating thermostat was installed at the 5 ft level on the west wall of the dining room, wired through a relay to the motors driving

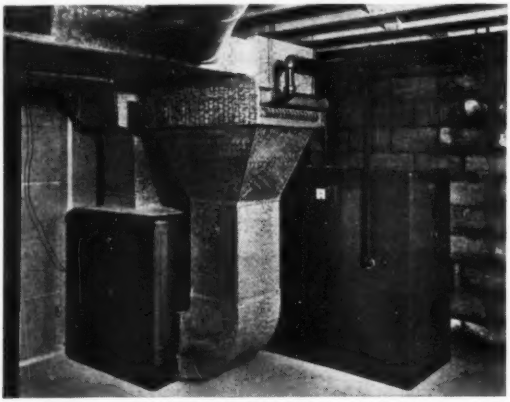


FIG. 3. VIEW OF BOILER, BLOWER, AND HEAT EXCHANGER

the blower and water circulator. The gas burner was controlled by an aquastat immersed in the water of the boiler.

Upon a call for heat by the thermostat the blower and circulator started simultaneously, delivering the warmed air to the panels, and on a 10 F drop in water temperature within the boiler the gas burner started, continuing to run until the 10 F had been restored. Upon satisfaction of the thermostat the blower and circulator stopped.

INSTRUMENTATION

During the construction of the house 176 24-gage copper constantan thermocouples were installed as follows:

21 in seven groups of three each for exposed walls gradients at 6 ft and 16 ft levels above grade on the south, east, and north walls and at the 16 ft level of the west walls one of each group embedded in the plaster; one on the outer face of the insulation and one embedded in the siding.

33 in 11 groups of three each for ambient readings located at different levels for different tests. One 3 in. or 6 in. above the floor, one at the 30 in. or 60 in. level and one 3 in. or 6 in. below the ceiling in each room except the lavatory. No thermocouples were shielded.

50 in 10 groups of five each embedded in the ceiling plaster for heating panel readings. One 15 in. from each corner of each panel and one in the center of each panel.

45 in nine groups of five each embedded in the finish floor for floor readings. One 15 in. from each corner of the room and one in the center of the room. No floor thermocouples were installed in the northeast bedroom.

24 in 12 groups, one at the entrance to and one at the discharge from each panel for readings of the inlet and discharge air temperatures.

3 embedded in the plaster of the north wall of living room at the 5 ft level, one opposite a supply riser, one opposite a return riser and one with no riser behind.

All thermocouples were connected to a central switchboard and potentiometer located in the basement (Fig. 4). A four point recording thermometer was used

for continuius records of ambient readings in the three bedrooms and bath on the second floor.

TEST OBSERVATIONS

Control System

When the house was erected it was anticipated that, with intermittent operation of the heating system, the thermal storage capacity of the wire lath and ¾ in. plaster comprising the ceiling panels might be of a magnitude that would result in objectionable temperature fluctuations within the living quarters. To determine this, observations were made over a period of 12 hours with the thermostat set at 70 F and with readings taken at the 5 ft level every 30 min, and 6 in. above the floor and 6 in. below the ceiling every two hours in the ten rooms of the first and second floors. The outdoor temperature averaged 12 F.

During the 12-hour period there were 26 cycles of operation totaling 3 hours 49 min. The average of the *on* periods was 8.8 min and of the *off* periods, 17.6 min.

The average of all readings 6 in. above the floor (Fig. 5) was 67.5 F and the maximum variation below this was 0.6 F and above, 0.4 F. The average of all readings at the 5 ft level was 70.7 F and the maximum variation below this was 0.7 F and above, 0.4 F. The average of all readings 6 in. below the ceiling was 72.0 F and the maximum variation below this was 0.4 F and above, 0.3 F.

As these observations did not indicate any objectionable temperature fluctuations, and because of the physical impossibility of recording readings frequently enough at all stations to obtain the temperatures at the exact times the thermostat called for heat and became satisfied, a test was made at a 30 F outdoor temperature of one room only. Readings were taken in the living

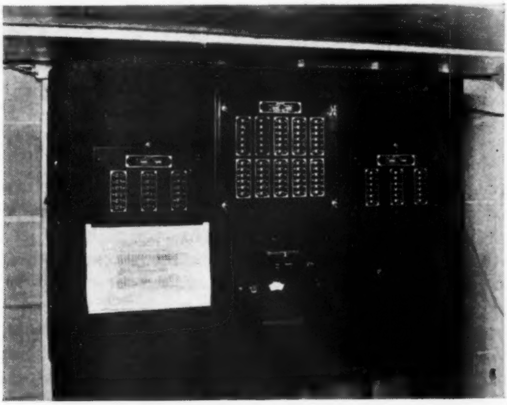


FIG. 4. VIEW OF SWITCHBOARD AND POTENTIOMETER

room 3 in. and 30 in. above the floor and 3 in. below the ceiling at the moment the thermostat called for heat and blower started and at the moment of thermostat satisfaction when the blower stopped. The test was conducted through a period of five cycles of system operation from 10 p.m. to 2 a.m. The average of the *on* periods was 7.3 min and the average of the *off* periods 33 min.

During the test period the greatest temperature variation at any one point was 1 F and the average temperatures at the beginning of the *on* period were 0.2 F higher than at the end of the *on* period. This result, being the reverse of expectancy, indicated the thermostat was affected by radiation and stopped the source of heat before the thermocouples registered a temperature rise and

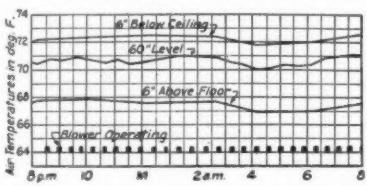


FIG. 5. GRAPHIC LOG OF AVERAGES TEMPERATURES AT THREE LEVELS WITH INTERMITTENT OPERATION. OUTDOOR TEMPERATURE, 12 F

that subsequent heat dissipation from the panel resulted in an increase in air temperature during the *off* period.

To compare this intermittent operation with the possible over-heating effect of a prolonged period of operation the house was cooled, on a day when the outdoor temperature was 16 F, to an average ceiling temperature of 60 F, floor of 59 F, inside wall 56 F, exposed wall 54 F and 30 in. level of 55 F.

At 12:20 a.m. the heating system was started and continued in operation for 4 hours and 25 min before the thermostat became satisfied. During that time the average temperature of the air leaving the exchanger in the basement was 130 F, entering the panels 123 F and leaving the panels 85.4 F, these latter two figures being the average of 120 readings. The air temperature drop per lineal foot of duct averaged 0.22 F, and the air temperature drop through the panels averaged 37.6 F.

For a period of 50 min after thermostat satisfaction the temperature of the air and all surfaces of the room continued to rise with the exception of the ceiling which cooled rapidly. In Fig. 6 these temperatures and the calculated British equivalent temperature are plotted against time. During the 50-min temperature rise subsequent to the heat source being stopped the calculated British equivalent temperature rose only 1.8 F and during the succeeding 60 min fell to a value closely approximating that at which the thermostat became satisfied and at which heat was again required. Air temperatures at the 3 in. and 30 in. levels were higher at 6:35, at the beginning of an *on* period than at 4:45, the beginning of the *off* period.

The temperature variation during all of these periods of observation was such that it was not deemed necessary to use a more elaborate control system so as to provide constant operation with the temperature of the heating medium regulated inversely with the outdoor temperature.

PANEL TEMPERATURES

Examples of calculating radiant heating are given in A.S.H.V.E. Guide[1] which indicate agreement between two different methods. In one example the panel area and temperature required to maintain a 71 F mean radiant tem-

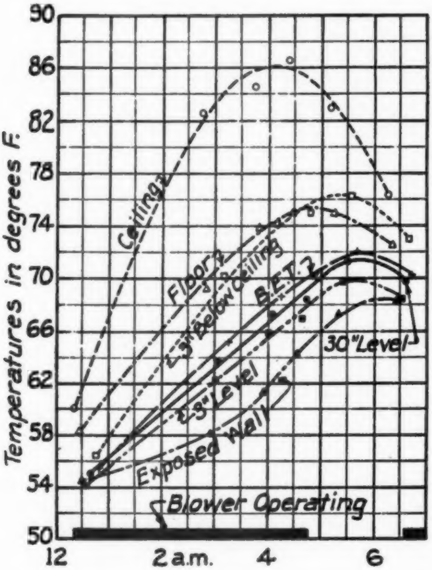


FIG. 6. SURFACE AND AIR TEMPERATURE CURVES DURING PERIOD OF HEATING AFTER HOUSE HAD BEEN COOLED

perature is computed from the difference between the total radiation of all interior surfaces at 71 F, and the total radiation of such surfaces at their calculated or assumed temperatures, with an indoor temperature of 65 F and 0 F outdoors. In another example the heat loss of the same room is computed in the conventional manner according to THE GUIDE 1943, Chapter 6, Heating Load. The British thermal unit requirements in both cases are approximately the same.

If the room used in these examples were partitioned into two rooms the British thermal unit requirements in the first example would be increased by

[1] HEATING VENTILATING AIR CONDITIONING GUIDE, 1943, Chapter 45, Radiant Heating.

the amount necessary to compensate for the difference in radiation from both sides of this partition at its desired temperature of 71 F and assumed temperature of 60 F. There would be no increase in the British thermal unit requirements computed by the second method as the partition would create no additional heat loss.

Applying the first method of calculation to this particular house the total heat emission from all interior surfaces, exclusive of the game room, as shown in Table 1, was 777,717 Btu per hour, equivalent to a mean radiant temperature of 61.8 F with emissivity of 0.94.

The inside surface temperatures were developed as follows: Outside wall glass and exposed floor from Fig. 7, p. 810 of THE GUIDE 1943. Inside wall was assumed to be no lower than the exposed wall, although footnote with

TABLE 1—HOUSE DATA FOR CALCULATING PANEL TEMPERATURE BASED ON 65 INSIDE AIR; 71 MRT; MPH WIND; −10 F OUTSIDE TEMPERATURE

SURFACE	AREA SQ FT	U	ESTIMATED INSIDE SURFACE TEMPERATURE DEG F	EMISSIVITY ϵ	HEAT EMISSION BTU PER SQ FT PER HR	TOTAL HEAT EMISSION BTU PER HR
Outside Wall	1254	0.083	62	0.95	121.4	152,236
Glass, Single	48	1.13	16	0.90	79.6	3,821
Glass, Double	262	0.75	32	0.90	90.9	23,816
Inside Wall	2058	...	62	0.95	121.4	249,841
Unexposed Ceiling	641	...	67	0.95	125.9	80,702
Exposed Ceiling	848	0.065	67	0.95	125.9	106,763
Exposed Floor	118	0.294	52	0.93	110.1	12,992
Unexposed Floor	1260	...	60	0.93	117.1	147,546
Total	6489			Avg. 0.94		777,717

Table 3, p. 812 of THE GUIDE 1943 assumes a temperature for such a surface as 60 F which assumed temperature was used for the unexposed floor. The estimated temperature of the exposed ceiling from Fig. 7, referred to, would be approximately $63\text{ F} \times 1.05 = 67\text{ F}$ and the temperature of the unexposed ceiling was assumed at the same value.

For a mean radiant temperature of 71 F, having a heat emission of 128.3, the total emission for all surfaces would be $6489 \times 128.3 = 832{,}539$ Btu per hour or an additional requirement of $(832{,}539 - 777{,}717)$ 54,822 Btu per hour. Dividing 54,822 by the panel area of 1267 gives 43.3 Btu per square foot per hour which, added to 125.9 Btu per hour, the emission of the unheated ceiling, equals 169.2 Btu per hour requiring a panel temperature of approximately 107.1 F.

Computing the heat loss of the house, exclusive of the game room, according to Chapter 6, of THE GUIDE 1943, including infiltration but excluding heat loss upwards from panels, the British thermal unit loss per hour at an 80 F temperature difference was, as shown in Table 2, 39,984. Dividing this by the panel area of 1267 gives 31.6 Btu per square foot per hour. As approximately 70 per cent of the heat from a horizontal panel with heat flow downward is by radiation, the Btu per square foot would be $31.6 \times 0.70 = 22.1$ which,

added to 129.6, the radiation from a surface at 71 F (the desired mean radiant temperature) with an emissivity of 0.95 gives a total radiation of 151.7 Btu equal to a panel temperature of 91.6 F. This temperature agrees with the average for the house when each room is calculated individually and it is this temperature that is used in the following comparison with test results.

To compare the calculated panel temperatures with actual requirements tests were run at various outdoor temperatures. For these tests the water temperature in the boiler was reduced to a point that required constant operation

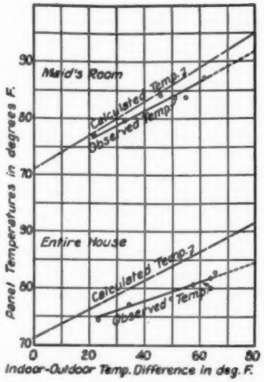


FIG. 7. CURVES OF OBSERVED AND CALCULATED PANEL TEMPERATURES

of the heating system. Readings were taken at all stations and each test was run over a sufficient period of time (generally 5 to 6 hours) to insure temperatures throughout the house being stabilized as indicated by two successive readings at 60-min intervals being in close agreement. In Fig. 7 the panel

TABLE 2—HEAT LOSS CALCULATION, NOT INCLUDING HEAT LOSS UPWARD FROM PANELS

SURFACE	AREA SQ FT	U	TEMPERATURE DIFFERENCE	BTU
Outside Wall	1254 ×	0.083 ×	80	8327
Glass, Single	48 ×	1.13 ×	80	4339
Glass with Storm Sash	262 ×	0.75 ×	80	15720
Exposed Ceiling, No Panel over	111 ×	0.065 ×	80	577
Exposed Floor	118 ×	0.294 ×	40	1387
				30350
Infiltration ... 283.5 ft[a] × 23.6 cfhr × 0.018 × 80				9634
				39984 Total

[a] 100 per cent of crack of windows, 200 per cent of crack of doors on north and east exposures.

temperatures are compared with the calculated requirements. Each point plotted represents the sum of the products of the average of 5 points in each room panel multiplied by its area, divided by the total panel area.

Extrapolating the observed temperature curve to the 80 F temperature difference gives a required panel temperature of 85 F which is 6.6 F below the calculated temperature of 91.6 F or 68 per cent of the calculator rise above 71 F based on the heat loss method of computation and 38.8 per cent of the calculated rise above 71 F based on the radiation method of computation. This discrepancy between the observed panel temperature and that calculated by the heat loss method might be accounted for by stray heat loss from the heating system and chimney. At design temperatures with an 0.25 F drop per lineal foot of ducts to panels, the heat dissipation from the supply side of the heating system alone would approximate 16 per cent of the calculated requirements. If the heater, return system and chimney accounted for a similar amount, and if the required heat input from the panels were computed less this total allowance, then the observed and calculated panel temperatures would be in close agreement.

The calculated average panel temperature for the house was 91.6 F but for the maid's room only it was 95 F. Into this room there was less stray heat than into any other room, as it was unaffected by ducts or risers other than to itself. Fig. 7 also compares the panel calculated temperatures with the test results in this room only where the recorded panel temperature rise above 71 F was 88 per cent of the calculated requirement.

TEMPERATURE GRADIENT IN LIVING ROOM

Observations of the gradient in the living room were made on two occasions, one week apart, with outdoor temperatures of 30 F and 30.5 F. One of the thermocouples attached to the standard reaching from floor to ceiling in the center of the room was moved in 3 in. increments in both directions between floor and ceiling and the temperature at each station recorded. In that way four sets of readings were taken over a total expired time of 6 hours. The average of all these readings, plotted on Fig. 8, is 71.48 F. The average of stations 3 in. and 30 in. above the floor and 3 in. below the ceiling is 71.5 F.

The temperature variation in the living room between 3 in. above the floor and 3 in. below the ceiling was of the order of 5.5 F.

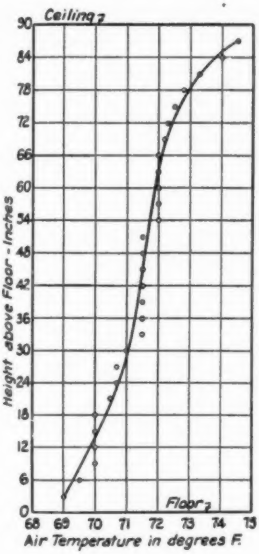
AIR TEMPERATURES COMPARED WITH MEAN RADIANT TEMPERATURES

As no means of measuring mean radiant temperature was used, it could only be arrived at from readings of 105 thermocouple locations in floors, walls and ceilings, and calculating the glass temperatures from heat transmission coefficients, and assuming partition surface temperatures to be the average of air temperature readings. The average temperature of these inner partitions, with the system of heating employed, probably exceeded the average air temperature due to heat transmission from riser pipes within the partition. Three thermocouples located on the north wall of the living room indicated an average surface temperature of 1.8 F above air temperature with no riser behind, 4.3 F higher with a return riser behind and 7 F higher with a supply riser

behind. If this is indicative of all the partitions, the mean radiant temperature reported herein is slightly lower than that which actually existed.

F. C. Houghten and others have shown[3] that in a room heated by free standing radiation the mean radiant temperature never exceeded the air temperature and readings plotted in Fig. 9 indicate a similar condition with this particular application of ceiling panel warming.

The curves in Fig. 9 showing temperatures of the floor, 3 in. above the floor, 30 in. above the floor, 3 in. below the ceiling, the ceiling, exposed wall



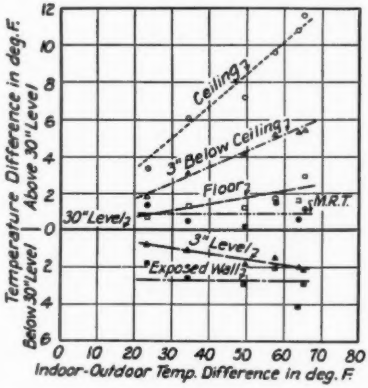


FIG. 9. SURFACE, AIR, AND MRT CURVES

FIG. 8 (*left*). TEMPERATURE GRADIENT IN LIVING ROOM. OUTDOOR TEMPERATURE 30 F

and calculated mean radiant temperature are plotted from tests run at outdoor temperatures ranging from 4 F to 47 F with the heating system under constant operation for a sufficient period of time to reach equilibrium as indicated by two successive readings at 60-min intervals. The calculated average mean radiant temperature for all tests exceeded the temperature at the 30 in. level by 0.9 F and likewise the average of the air temperatures 3 in. and 30 in. above the floor and 3 in. below the ceiling was 0.9 F above the 30 in. level. If the mean radiant temperature were compared only with the air temperature below the breathing line then the mean radiant temperature would be of a slightly higher value.

This system employed horizontal panels with heat flow downward from which heat transmission by convection is less than from vertical panels or from horizontal panels with heat flow upward. It is doubtful, therefore, in any well constructed panel heated house having no more than a normal number of air

[3] A.S.H.V.E. Research Report No. 1172—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst and S. Suciu, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 117.)

changes that the air temperatures would ever be depressed appreciably below the mean radiant temperature.

With radiator and convection residential heating systems the gradient between the 3 in. and 30 in. levels has been found to increase with an increase in the indoor-outdoor temperature differential.[3] While a similar condition existed in this investigation the floor temperatures, as shown in Fig. 9, not only exceeded the air at the 30 in. level but also the floor temperature increased with the indoor-outdoor differential.

As a drop in outdoor temperature was accompanied by an increase in ceiling and floor temperatures, a rather constant exposed wall temperature and a decreased glass temperature, the mean radiant temperature held within such

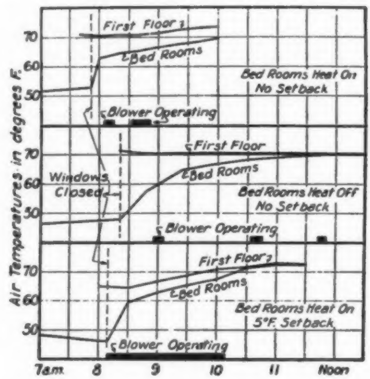


Fig. 10. Graphic Log of Temperatures at 30 In. Level During Morning Pick-up

narrow limits that comfort conditions were maintained during all outdoor conditions without a change in thermostat setting.

Morning Pick-Up

Three tests were run to observe conditions existing in the morning after bedroom windows, which had been opened during the night, were closed. In the first test the heat remained on in the bedrooms, the thermostat was not set back, the bedroom doors were closed and windows were opened 3 in. from the top and 6 in. from the bottom. The second test was made under similar conditions except that heat in the bedrooms was turned off. In the third test heat remained on in the bedrooms but the thermostat was reduced 5 F during the night. In all three tests the outdoor temperature ranged from 30 F to 34 F. These three tests are plotted in Fig. 10.

In the first test with heat remaining on in the bedrooms and no night

[3] Operation of the Research Home with Reduced Room Temperatures at Night, by A. P. Kratz, W. S. Harris and M. K. Fahnestock. Digest of Research, Engrg. Experiment Station, University of Illinois. (A.S.H.V.E. Transactions, Vol. 49, 1943.)

set-back, the bedroom temperatures dropped to an average low of 51 F. At 7:52 a.m. the bedroom doors were opened and windows closed. On demand from the thermostat the heating system operated from 8:04 to 8:15 and from 8:32 to 8:52 at which time the second floor bedrooms were 5 F lower than the first floor. The heating system did not operate again for several hours due to sun effect.

In the second test, in which heat in bedrooms was turned off but with no set-back, the bedrooms reached a low average temperature of 45 F. At 8:30 a.m. the bedroom doors were opened, windows closed and dampers opened in the risers. On demand from the thermostat the heating system operated from 8:55 to 9:05—from 10:33 to 10:45 and from 11:41 to 11:50 or a total of 31 min out of 3 hours and 30 min. At 11:50 a.m. the temperatures of bedrooms had equalized with the first floor.

Fig. 11 is a reproduction of the chart from the 4 point recorder from 7:00 a.m. to 1:00 p.m. Temperature of the bathroom held within 1 F of 70 F until 8:30 a.m. when the bedroom doors were opened. The bathroom tem-

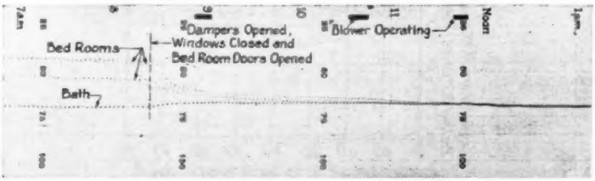


FIG. 11. REPRODUCTION OF FOUR-POINT RECORDER CHART COMPARING TEMPERATURE AT 30 IN. LEVEL IN BATH WITH THREE BEDROOMS IN WHICH WINDOWS WERE OPEN DURING NIGHT AND HEAT OFF

perature then dropped for about one hour while the bedroom temperatures rose. All of them equalized at 11:50 and reached and continued at 70 F at 12:30 p.m.

In the third test heat remained on in the bedrooms but the thermostat was reduced 5 F. During the night the mean bedroom temperature dropped to 46 F. The doors were opened, windows closed and thermostat advanced 5 F at 8:08 a.m. at which time the heating system started and continued in operation until 10:08 a.m. when the temperatures on both floors were practically equalized. For an hour thereafter temperatures continued to rise—due partially, at least, to solar radiation. Under normal operation, however, the thermostat would have been automatically advanced at least one hour before the bedroom doors were opened so that any overheating would be minimized.

EFFECT OF LARGE GLASS AREAS

Exposure of the dining room consisted of 115 sq ft of wall and 77.3 sq ft of glass and doors, the windows and french doors taking up 12 lineal feet of wall at the floor line.

A comparison of floor temperatures and air temperatures 3 in. above the floor in the dining room and the average of the house is shown in Fig. 12.

These curves were plotted from readings of constant operation tests at various outdoor temperatures.

The variation between the 3 in. and 30 in. levels in the dining room was 137 per cent greater than in the house as a whole due apparently to the convection currents established by the large glass area. Because the floor was warmer than the air at either the 3 in. or 30 in. level, the lower air temperature was not of a magnitude to seriously affect comfort but it does suggest the same consideration for glass placement with panel heating as with any other.

SUMMARY

1. A basement game room from which the heat loss fluctuation was not uniform with the balance of the house, could not be satisfactorily heated in mild weather.

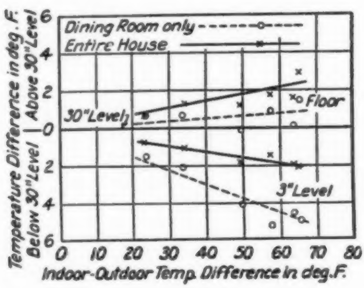


FIG. 12. CURVES OF FLOOR AND AIR TEMPERATURE AT 3 IN. LEVEL IN DINING ROOM AND ENTIRE HOUSE

2. The combined thermal capacity of air as the heating medium and normal plaster ceilings as the heating panels resulted in such rapid response to heat demand that comfort conditions were maintained with intermittent operation of the heating system under control of a conventional heat anticipating thermostat.

3. In this house with normal air changes, and warmed by a panel heating system, the air temperature tended to equalize with the mean radiant temperature.

4. Calculated panel temperatures based on the heat loss of the structure more closely approached observed requirements than the calculated temperatures based on radiation.

5. Stray heat loss into the house from chimney, heater and duct work provided a substantial factor of safety.

6. The mean radiant temperature was held within such narrow limits over a wide range of outdoor temperatures that comfort conditions were maintained without altering the indoor air temperature.

7. After a reduced night temperature the extended period of heater operation, required to restore day time temperature, reduced the time required for temperature equalization of rooms, in which windows were opened during the night, with the remainder of the house.

8. Floors above the basement increased in temperature as the outdoor temperature dropped and temperature of such floors exceeded the temperature of air adjacent to them. This higher floor temperature tended to counteract the affect of convection currents and depressed low level air temperatures established by large exposed glass areas.

DISCUSSION

T. Napier Adlam, New York, N. Y. (Written): It appears from the authors' statement that the instrument used to record the air temperatures did not indicate any great change over a given period—indicating that the instruments used were not capable of registering comfort conditions.

The air temperatures recorded being very much higher than necessary with a true radiant heating system would disturb the balance of heat loss from the human body. While the calculations called for a panel surface temperature 107.1 F, the average surface temperatures of the panels were probably 91.6 or 95 F. This can be accounted for by the extra heat coming from the floors and warmed portion of the walls, also by the fact that while the calculation made for the building was based on an outside temperature of −10 F, the test was made with an outside temperature of +12 F.

F. E. Giesecke, Urbana, Ill. (Written): The authors have made a valuable contribution to the panel heating literature. They have developed a panel heating system which is new and simple and have shown that, with this system, the temperature can be controlled satisfactorily with the conventional thermostat. They have also supplied experimental data which can be used for further studies of panel heating. For example, Fig. 9 shows that the MRT was about 1 deg above the 30 deg level air temperature.

This is evidence that, in any one room, the air temperature and the MRT are interdependent. If a room is heated by means of panels until its thermal conditions are constant, the MRT will be slightly higher than the mean air temperature, because the air is heated by its surroundings.

If a room is heated by means of a convector until its thermal conditions are constant, the MRT will be slightly lower than the mean air temperature, because the heat transfer is from the air to its surroundings.

In the case of the building studied by the authors, it is impossible to design the heating system so that there will be a definite MRT and also a definite mean air temperature unless the two selected temperatures happen to correspond to each other. For example, it is evident from the authors' experimental data that when the MRT was 71 F, the mean air temperature was about 70 F and could not have been reduced to 65 F, unless cool air was continuously blown into the room in sufficient quantity to lower the air temperature from 70 to 65 F.

In the authors' system, the ceiling panel was heated by blowing the air along the *upper* surface of the panel. It would be of interest and of value if the authors could continue their studies by heating the ceiling panel with air blown *across the room* along the *lower* surface of the panel. To secure design data for such an installation, tests were conducted in a radiator test booth at the University of Illinois. While a radiator was being tested, the ceiling temperature was about 93 F and the air temperature, 3 in. below the ceiling, was about 100 F, or 7 deg higher. After the radiator test had been completed, the test booth was closed and allowed to heat until the ceiling temperature was about 101 F; at the same time the air temperature was about 111 F, or 10 deg higher. The heat was then cut off and the booth allowed to cool until the ceiling temperature was about 95 F at which time the air temperature was about 100 F, or about 5 deg higher.

It appears from this test that a ceiling temperature of 86 F in the booth could be easily attained with an air temperature of 93 F, 3 in. below the ceiling.

In the authors' tests, it appears from Fig. 6 that a ceiling temperature of 86 F was secured by blowing air over the ceiling at a mean temperature of about $\left(\frac{125+85}{2}\right) = 105$ F, or about 12 deg higher than necessary if the air were floated along the underside of the ceiling panel. A system constructed and operated as suggested would be lower in cost and possibly also lower in the cost of operation

than the system used by the authors and, in addition, it would permit partial purification and partial humidification of the air, so that it might be classified as a combination panel heating and air conditioning system. I hope the authors will consider these suggestions sympathetically and present another paper in the future.

A. J. JOHNSON, Philadelphia, Pa. (WRITTEN): The research of the authors has undoubtedly added invaluably to our fund of domestic heating knowledge. The paper is so clearly presented and the subject so new that there seems to be little room for extended discussion at this time.

There is one point, however, which might warrant further consideration. The paper mentions the fact that it required 4 hours and 25 min to raise the temperature from 60 deg to a point where the thermostat was satisfied. It later refers to 2 hours of continuous heater operation for recovering 5 deg in the morning. Both of these acceleration periods would seem to be rather long, which suggests that with a low temperature, concentrated heating medium, increased attention might well be given to air circulation and thermostat placement.

A point that is not always realized is that an eight-room house contains over a ton of air, which must be placed in continuous motion in order to secure uniform heating. Since the only motive power after the fan has stopped is the source of heat supply, it follows that when this is either low in temperature or concentrated in area more time is required to overcome the inertia of the air than otherwise. This would account for delayed acceleration, over-riding in rooms adjacent to the thermostat as mentioned in this paper, and a tendency toward difficulty in balancing the temperatures in the several rooms.

Several suggestions for increasing the comfort in a house of this type are therefore indicated:

1. It might be advisable to study the relative fuel consumption with uniform 24-hour temperatures as compared with depressed night temperatures. If it can be shown, as might be suspected, that in any house with a system requiring prolonged acceleration periods, 24-hour comfort does not involve appreciable additional fuel, the increased comfort factor is obvious.

2. In designing a low temperature source of heating, it would seem to be in order to carefully study the path of convected currents through the house, so that each heating panel will in effect relay the normal flow of heated air to the next panel rather than setting up localized circulation or opposing currents.

3. A change in the location of the room thermostat might be in order. With intermittent types of heat there is a tendency for the room in which the thermostat is located to become heated before the circulation of air throughout the house is normal. Thus, instead of the design condition of each radiator or panel satisfying its own area there is necessarily a secondary effect, due to the cooler rooms having to draw upon the heated air in the warmer rooms before stabilization has been achieved. This condition can probably be overcome by a very painstaking balancing of the entire heating system. However, a simpler solution might be found if the thermostat could be located in point of average temperature such as for example an upstairs hall.

B. F. RABER and F. W. HUTCHINSON, Berkeley, Calif. (WRITTEN): This paper affords an interesting opportunity to check the accuracy of existing procedure for panel heating design. The so-called radiant method (A.S.H.V.E. GUIDE, 1943, Chapter 45) is shown by the authors to indicate a required panel temperature of 107.1 F. The standard convection method (THE GUIDE, Chapter 6) is shown to indicate a panel temperature of 91.6 F. The corresponding experimental panel temperature, by extrapolation, is found to be 85 F. Thus the so-called radiant method gives a heating system $\frac{107.1-70}{85-70}=247$ per cent as large as that which was actually required; the convection method gives a design $\frac{91.6-70}{85-70}=144$ per cent as great as actual (these values are exclusive of correction for stray heat losses from the system, as explained in the paper).

These results serve, in the writers' opinion, to provide overwhelming evidence to substantiate the frequently expressed belief that the radiant design procedure suggested

in THE GUIDE is unsound from the standpoint of practice (147 per cent *excess* heating plant!) as well as theoretically irrational. As Professor Giesecke has stated, air temperature and MRT are dependent variables and cannot, as is done in THE GUIDE method, be arbitrarily assigned values independent both of one another and of the characteristics of the structure.

In addition to giving an erroneous panel temperature, the method specified in Chapter 45 also indicates that the panel heating load is $247/144 = 172$ per cent as great as the equivalent convection heating load. If this were correct there would be little justification for use of panel heating; most published data, in this and other papers, indicate that it is not correct.

A third method of panel design, based on taking a heat balance on the unheated interior surface, has been developed by the writers[4] and used in correlating data from the panel heating program which is being conducted at the University of California as a cooperative research project with the A.S.H.V.E. This method is summarized in *Equations I, II,* and *III* in the appendix of this paper. Application of this heat balance method to the present case gives a calculated panel temperature of 87.6 F, or $\frac{87.6-70}{85-70} = 117$ per cent as great as the actual.

Thus the three methods indicate over-design for the heating plant by 147 per cent (Chapter 45), 44 per cent (Chapter 6) and 17 per cent (heat balance). Since convection heating from sources other than the panels was known to have contributed a substantial fraction of the heating requirements, the third method appears to agree, within the limits of accuracy of the coefficients used and of the experimental data, with the test results. Detailed calculations for the heat balance method are as follows:

Rational Panel Design Procedure by Heat Balance Method

1. Ventilation (from Table 2) $= 283.5 \times 23.6 = 6690$ cfh
2. Total enclosure surface (Table 1) $= 6489$
3. Ventilation rate, V_c, cubic feet per hour per square foot $= \frac{6690}{6489} = 1.03$
4. Equivalent overall coefficient, V, (Table 2) $= (30{,}350 + 1{,}387)/(6489)(80) = 0.061$
5. Equivalent conductance, $C_w, = \frac{1.65U}{1.65 - U} = \frac{1.65 \times 0.061}{1.65 - 0.061} = 0.0634$
6. Panel area (from text of paper) $= 1267$ sq ft
7. u = fraction of total surface heated $= 1267/6489 = 0.195$
8. v = fraction of total surface unheated $= 1 - u = 0.805$
9. Design outside temperature (from text of paper) $= -10$ F
10. Substituting in Equation I (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 435)

$$t_w = \frac{ut_p + 0.8vt_a + C_w vt_o}{u + 0.8v + C_w v} = \frac{0.195t_p + 0.8 \times 0.805t_a + (0.0634)(0.805)(-10)}{0.195 + 0.8 \times 0.805 + 0.0634 \times 0.805}$$
$$= 0.219t_p + 0.724t_a - 0.573$$

or $t_a = 1.38t_w - 0.302t_p + 0.792$

11. Substituting in Equation II (on page 436, A.S.H.V.E. TRANSACTIONS, Vol. 48) in which the first two terms in the denominator should be $0.4u$ and $0.8v$,

$$t_a = \frac{0.4ut_p + 0.8vt_w + 0.0178V_c t_o}{0.4u + 0.8v + 0.0178V_c} = \frac{0.4 \times 0.195t_p + 0.8 \times 0.805t_w + 0.0178 \times 1.03(-10)}{0.4 \times 0.195 + 0.8 \times 0.805 + 0.0178 \times 1.03}$$
$$= 0.1054t_p + 0.87t_w - 0.247$$

12. Revising Equation III for comfort temperature of 70 F,

$$t_a = 140 - ut_p - vt_w = 140 - 0.195t_p - 0.805\,t_w$$

13. Equating right-hand sides of expressions from steps 10 and 12,

$$1.38t_w - 0.302t_p + 0.791 = 140 - 0.195t_p - 0.805t_w$$
$$t_w = 63.7 + 0.049t_p$$

14. Equating right-hand sides of expressions from steps 11 and 12,

$$0.1054t_p + 0.87t_w - 0.247 = 140 - 0.195t_p - 0.805t_w$$
$$t_w = 83.7 - 0.1794t_p$$

15. Equating right-hand sides of expressions from steps 13 and 14,

$$63.7 + 0.049t_p = 83.7 - 0.1794t_p$$
$$t_p = 87.6 \text{ F}$$

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[4] Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson. A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942.

F. C. HOUGHTEN, Pittsburgh, Pa.: From a theoretical as well as pratical viewpoint the addition of a partition to a given space would increase the heat loss because the partition would take on a higher temperature than the exposed surface would have a higher mean radiant temperature, and would therefore, radiate more heat to the inside surfaces of the exposed wall. The A.S.H.V.E. GUIDE method of computing the radiant panel size is theoretically correct.

C. TASKER, Cleveland, Ohio: There is need to standardize the method of measuring radiant heat in order that field test results of different investigators may be comparable.

P. B. GORDON, Bloomfield, N. J.: How much insulation was used between the first floor ceiling and second floor construction, and has the heat flow through the second floor construction entered into the discrepancy between the observed and calculated mean radiant temperature? The observed ceiling panel temperature might be lower than the calculated value because the first or second floor construction temperature and the plaster surface temperature of the interior walls might have been higher than calculated due to conveyance of air up through ducts in the walls. It is probable that the air temperature could not be held below the mean radiant temperature because a lot of surface at points other than the ceiling was effective as radiant heating surface.

L. T. AVERY, Shaker Heights, Ohio: Was the relative humidity determined when the conclusions were reached regarding comfort?

JOHN JAMES, Cleveland, Ohio: How did the authors secure uniform distribution of air flow in the thin ceiling panels? Were special precautions taken to prevent discoloration or cracking of the plaster?

ELLIOTT HARRINGTON, Schenectady, N. Y.: I would be interested in the authors' impression of the comfort characteristics or operating advantages of panel heating.

FERDINAND JEHLE, Indianapolis, Ind.: Would warm air panel installations usually employ a forced warm air furnace and if so could the air temperature in the panel be controlled with sufficient accuracy?

C.-E. A. WINSLOW, New Haven, Conn.: This paper has shown two advantages that might be expected from radiant heating by a ceiling panel: (1) the differential between floor and ceiling was small as compared with most convection systems; and (2) there was less air movement.

Convection heat loss from the body depends on the mean surface temperature of the body and the mean air temperature multiplied by the square root of the air movement. At a mean velocity of 16 ft instead of 64, for instance, the difference between mean skin temperature and mean air temperature can be twice as great with the same sensation of comfort. The evenness of temperature from floor to ceiling and the low air movement permits use of low air temperature with the same comfort.

We have made successful use of banks of light bulbs against the ceiling and floor for the purpose of rapidly raising room temperature in the morning in the J. B. Pierce Laboratory. Room temperature can be raised from 55 to 70 F in 30 to 45 min by temporarily increasing the wattage input.

AUTHORS' CLOSURE: Referring to remarks by Messrs. Adlam and Tasker, no standard of measurement for registering comfort exists at the present time. With the increasing interest in radiant heating it would be well to have a definition of the term *true radiant* heating system. It is not clear at present whether this term refers to the means by which the heat is introduced into a room, or whether it refers to a relation between air temperature and the MRT.

Dr. Giesecke's suggestion of using warm air under the panel rather than over it, is interesting. I see no reason why it would not work. As the air would be

at a higher temperature than the panel, objectionable dust and dirt streaks would undoubtedly be formed on the ceiling panel.

In answering Mr. Johnson, who requested the time required to recover 5 F in the morning after a night setback, it was found that the rate of recovery could be shortened by using a slightly higher temperature in the heating medium, but the elapse of time shown in the paper is no greater than that experienced in warm air and radiator heated homes at the University of Illinois. This recovery refers only to the air temperature and occupants of panel heated homes apparently experience comfort conditions soon after the heating system is in operation and before any appreciable air temperature elevation has been noticed.

Commenting on Commander Houghten's remarks it is agreed that by running an internal partition in a room the parition would take on a higher temperature than the exposed surface, thereby increasing the MRT, but we do not understand why a greater heat input would be required which is the case when calculated according to Chapter 45 in the A.S.H.V.E. GUIDE. If, in a room 20 ft x 24 ft with the north and south walls exposed and having areas equal to those given in THE GUIDE example, a partition is run from north to south the required heat input would be increased some 15 per cent. Because of such a condition we believe the heat loss method of calculation should be used.

Mr. Gordon is correct in his assumption that floor temperatures were increased as a result of panels or duct work underneath, and likewise as pointed out in the paper partition temperatures were elevated by risers therein. It did not seem to the authors as though the discrepancy between calculated and observed panel temperatures was great when the heat dissipation of the supply side of the system alone accounted for 16 per cent of the heat loss, which was not considered in calculating the panel temperatures. It may be that if heat input had been obtained from the ceilings exclusively the air temperature in relation to the MRT would have been somewhat lower; however, it is the authors' opinion that in an ordinary residence, without ventilation and using low temperature panels, the air will eventually approach the MRT.

Six or seven installations of the type described in the paper had been made and all were equipped with direct fired units either gas, oil or stoker-fired and all have been very satisfactory to the owners. There is no intention to conclude that the warm air panel system was better than a well installed radiator or convection warm air heating system in a well-built house. The ability to place draperies or furniture where desired without need for considering the heating system is an admitted advantage.

No cracking of plaster has been observed in the installations made. Three made use of expanded lath with conventional plaster (without binding) and two or three made use of gypsum lath and plaster attached to it. No discoloration had been observed although the panels had been painted to match the rooms. In the installation described the plaster was smooth. Humidity was not determined because the house was unoccupied, although its bearing on comfort was acknowledged. Radiant temperatures were not obtained because the method of taking them had apparently not been standardized.

In comparing the time of morning temperature pickup with that reported in tests of a radiator system in the Research Residence at Urbana two hours were required in the latter to recover 6 F and cold walls at the end of this period still produced an uncomfortable effect on the occupants.

No. 1237

USE OF THE DOWN-DRAFT COKING METHOD FOR SMOKELESS COMBUSTION

By JULIAN R. FELLOWS * AND JOHN C. MILES,** URBANA, ILL.

INTRODUCTION

THE well known fact that the temperature required to ignite a coal gas-air mixture is much higher than that required to convert the volatile matter in bituminous coal to the gaseous state, makes it necessary to use either the down-draft principle or the underfeed principle in the construction of small furnaces designed to achieve smokeless combustion. The underfeed principle has been used to some extent in hand fired furnaces but its use has been generally abandoned because the operation of a furnace equipped with the necessary mechanism for forcing the coal through the bottom of the fuel bed was found to be too difficult for the average housewife to operate. The down-draft principle has also been tried in innumerable forms by many inventors and furnace designers, but for various reasons, such as grate failure, baffle wall failure, arching of the coke, difficulty in removing clinkers and "puff back," practically every attempt to use this principle has resulted in ultimate failure.

A project of the Mechanical Engineering Experiment Station at the University of Illinois started in 1939 by the authors, with the cooperation of Prof. A. P. Kratz, has had for its objective the practicable application of the down-draft principle to hand fired furnaces designed to burn bituminous coal smokelessly. In developing the down-draft coking principle an attempt has been made to eliminate all of the faults that have characterized previous applications of the down-draft principle.

THE DOWN-DRAFT COKING PRINCIPLE

Fig. 1 shows a longitudinal vertical section of the body of a furnace incorporating the down-draft coking principle. This figure illustrates the method of burning the fuel, without being concerned either with the arrangement of secondary heating surface or with the use that is made of the heat liberated during the combustion process. The typical furnace consists of a coking chamber, a coke burning chamber, a baffle wall with vertical air passages, a sloping dead-plate under the coking chamber, an inclined "pin hole" grate under the forward portion of the coke burning chamber and a horizontal shaking grate under the rearward portion. The baffle wall containing vertical air passages separates the upper portion of the coking chamber from the combustion chamber and directs the gases released from the coking coal over the surface of the incandescent coke in the coke burning chamber. Three definitely pro-

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portioned openings admit air in the correct proportions to the secondary air passages, the coking chamber, and the ash compartment below the grates. The secondary air which enters the furnace through ports above the firing door passes through a horizontal passage in the roof of the coking chamber, then through the vertical passages in the baffle wall, emerging from the bottom of the baffle wall, where it is thoroughly mixed with the gases passing toward the combustion chamber from the coking chamber. The primary coking air, in an amount that is just sufficient to support a low combustion rate, enters the coking chamber through a small port in the firing door. The heat liberated by the slow combustion in the coking chamber gradually drives the volatile

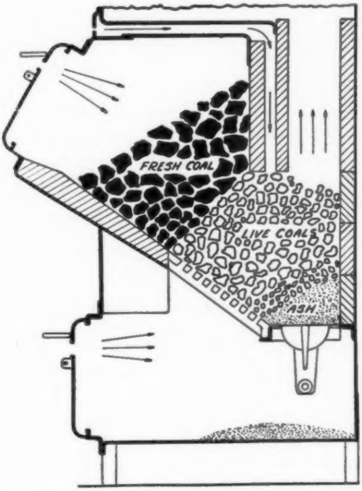

FIG. 1. SECTIONAL VIEW OF THE BODY OF A FURNACE INCORPORATING THE DOWN-DRAFT COKING PRINCIPLE

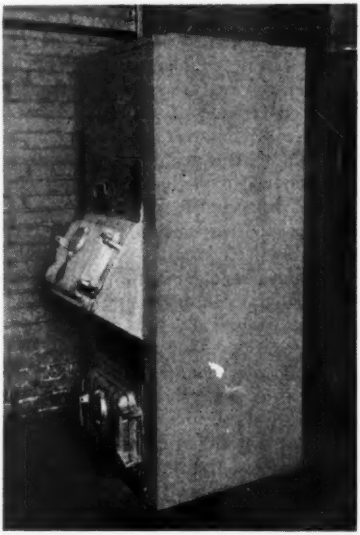

FIG. 2. PHOTOGRAPH OF A SMALL WARM AIR FURNACE INCORPORATING THE DOWN-DRAFT COKING PRINCIPLE

matter from the fresh coal, thus converting the coal to coke. The primary under grate air which enters the furnace through a small port in the ash removal door passes upward through the two portions of the grate and flows through the bed of coke. The primary coking air is proportioned to the primary under grate air in such a way that a charge of fresh coal is completely coked in the time interval required to entirely burn a charge of coke. The primary coking air, primary under grate air, and secondary air are admitted to the furnace in fixed proportions at all times, and the combustion rate is controlled by a damper which regulates the effective draft at the smoke collar.

METHODS ADOPTED TO IMPROVE DOWN-DRAFT PRINCIPLE

The application of the down-draft principle is not required for the smokeless combustion of the coke which remains after the hydrocarbon gases have been

removed. Hence for the purpose of burning the fixed carbon in the fuel, an updraft firepot is provided at the rear of the down-draft coking furnace. This arrangement makes it possible for the grate to be cooled by the primary air, as it is in any conventional updraft furnace. The grate used is made in two parts. A conventional shaking grate, used for disposing of the ash, is preceded by a sloping pin-hole grate. This arrangement makes it possible to vary the effective grate area by accumulating different amounts of ash at the back of the fire-pot. Since the forward portion of the inclined grate is always clear after a charge of fuel has been placed, the fire responds immediately regardless of the size of the coke bank. Hence, even in very mild weather, when small charges of coal are used and the fire is allowed to burn for long periods without attention, it is possible to maintain an ignition surface under the forward portion of the baffle wall.

The coking chamber at the front of the furnace is provided with a deadplate inclined toward the coke burning chamber at the back so that gravity aids in the transference of the coke from the coking chamber to the coke burning chamber. However, the angle of inclination is such that the coke does not feed on to the grate automatically but remains in the coking chamber until transferred to the coke burning chamber by poking, previous to the placing of a fresh charge of coal. With this arrangement sufficient coke for the ignition of the gases from the succeeding charge may be retained in the furnace for long periods.

Because only a small portion of the air entering the furnace is admitted through the coking chamber, it is impossible for the fuel in this chamber to become excessively hot, and the baffle wall is not subjected to extremely high temperatures. The resistance to the flow of air through the small port in the firing door is such that the resistance of the charge of fresh coal in the coking chamber is negligible, and, as a result, the rate of coking is independent of the preparation of the fuel used, providing that fine dust has been removed. The introduction of the secondary air through the baffle wall combines the advantage of cooling the baffle wall with that of preheating the secondary air.

While the coke formed in the coking chamber will bridge the space below the baffle wall, this condition causes no trouble in a down-draft coking furnace, because it does not affect the burning of the coke in the updraft fire pot below the baffle, and the various parts of the furnace are arranged so that the coke can be easily broken up and pushed into the coke burning chamber at the end of a cycle.

Clinkering of the ash is eliminated because most of the coke is burned on the inclined *pin-hole* grate which uniformly distributes the primary under grate air, and because the port which admits this air to the ash pit is designed to limit the amount at all times.

The tendency toward *puff-back* is reduced by using a baffle that is much shorter than those used in most of the previous applications of the down-draft principle. *Puff-back* can be definitely eliminated in the down-draft coking furnace by using the proper proportion of primary coking air.

APPLICATION TO A WARM AIR FURNACE

Fig. 2 shows a small warm air furnace designed to operate on the down-draft coking principle, and which is now being tested in the Mechanical Engineering

Laboratory at the University of Illinois. The furnace shown was developed by W. D. Redrup, Huntington, Ind., with the cooperation of Prof. A. P. Kratz, Prof. S. Konzo and the authors of this paper. This unit is 18 in. square and is fitted with a casing 26 in. by 26 in. It will be rated at approximately 60,000 Btu per hour with forced-circulation of the air. The same unit may be used with gravity-circulation, or as a circulating space heater. This furnace is so designed that it can be made in any desired capacity by maintaining approximately the same geometric proportions used in the unit now being tested.

PERFORMANCE DATA FOR A DOWN-DRAFT COKING FURNACE

Some data on the performance of the original laboratory model in burning bituminous coal from 8 different fields are given in a previous[1] paper. The

TABLE 1—PERFORMANCE DATA FOR A DOWN-DRAFT COKING FURNACE BURNING SALINE COUNTY, ILL., COAL

Average combustion rate, lb per hr	1.5	3	5	7.5[a]
Length of cycle, hours	24	12	8	6
Weight of each charge fired, lb	36	36	40	45
Average draft at smoke collar, in. H_2O	0.005	0.017	0.04	0.06
Average CO_2 in flue gas, per cent	9.4	8.72	10.56	11.47
Average flue gas temperature, deg F	224	333	526	600[a]
Average heat exchanger temperature (Fire pot level)	390	613	781	725[a]
Average heat exchanger temperature (Top of combustion chamber)	324	456	726	590[a]
Average temperature of heated air	126	150	209	137[a]
Average temperature of air entering bonnet	62.3	70.0	67.5	64
Average smoke density, per cent (Corrected for zero reading of recorder)	0.6	2.0	4.0	1.0
Heating value of coal as fired, Btu per lb	12,880	12,880	12,880	12,880
Total stack losses, Btu per lb of fuel	1,217	1,748	2,458	2,810
Grate loss, Btu per lb of fuel	847	847	847	847
Total heat utilized, Btu per lb of fuel	10,731	10,285	9.475	9,223
Total heat utilized, Btu per hour	16,224	30,855	48,375	69,172[a]
Overall efficiency, per cent	84	79.8	73.6	71.6

[a] Forced circulation of air being heated.

performance of the furnace shown in Fig. 2 when tested under conditions designed to simulate cold weather operation, mild weather operation and night banking, is also shown in graphical form in a recently published paper.[2] Table 1 presents additional, and more recent, data taken at 4 different burning rates when burning 2 in. x 3 in. nut coal from Saline County, Ill. The proximate analysis of the coal as fired was as follows: moisture 3.5 per cent, volatile matter 38.22 per cent, fixed carbon 51.61 per cent and ash 6.67 per cent.

Fig. 3 shows in graphical form some of the more important items of the data presented in Table 1. The first 3 tests were made with gravity-circulation of the air through the casing without any attached pipes while in the last test at 7.5 lb combustion rate, approximately 700 cu ft of air was circulated through

[1] An Improved Hand-Fired Furnace for High Volatile Coals by J. R. Fellows and J. C. Miles. (A.S.M.E. *Transactions*, Vol. 64, page 161, April, 1942.)
[2] Performance Characteristics of a Down-Draft Coking Furnace by J. R. Fellows. (*Mechanical Engineering*, May, 1943.)

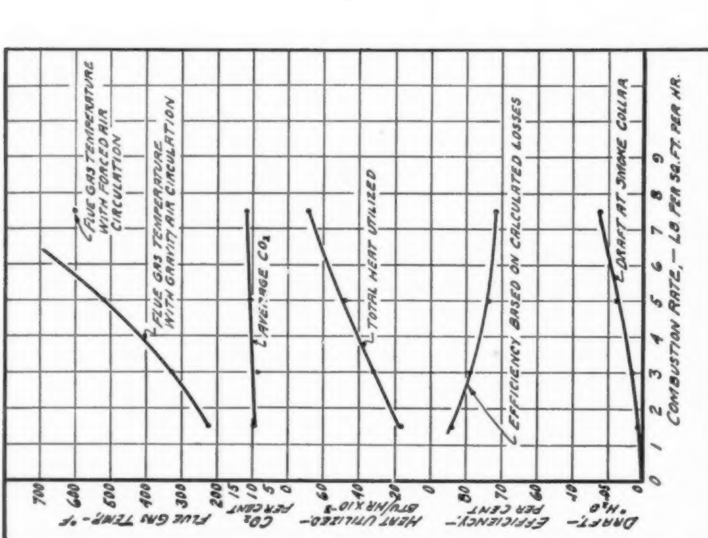

FIG. 3. PERFORMANCE CURVES FOR A WARM AIR FURNACE BURNING HIGH VOLATILE COAL

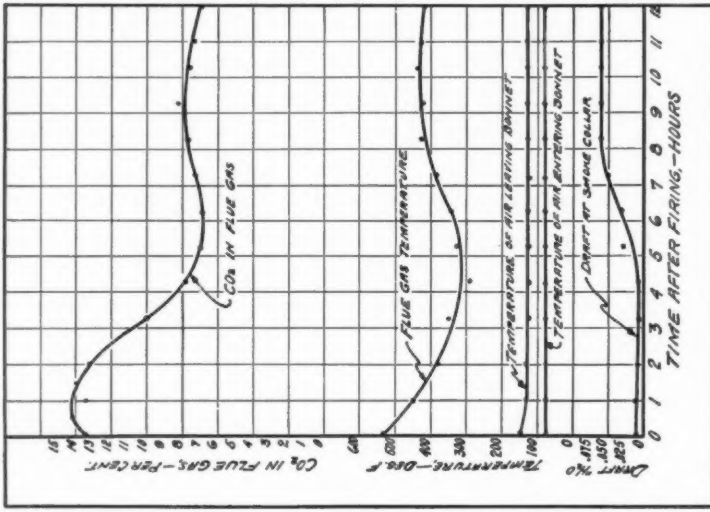

FIG. 4. GRAPHICAL LOG FOR A 12 HOUR CYCLE

the casing per minute by means of a fan. All of the tests were made without adjustment of any of the three air dampers and in all but the 24 hour test, at a 1.5 lb average combustion rate, the cross damper in the breeching remained in one fixed position through each test period. In the 24 hour test, the cross damper was moved to a *hold-fire* position after 11.5 hours of operation at a somewhat higher combustion rate. Other tests have shown that an average combustion rate as low as 1 lb per square foot of grate area per hour can be maintained without appreciable sacrifice in efficiency and with almost 100 per cent smokeless combustion. In every case the correct position of the cross damper for the combustion rate desired for a test was determined by preliminary trials. The draft in the breeching between the cross damper and the stack was maintained constant at 0.07 in. of water in all tests. The smoke density was measured by means of a recording meter making use of a light shining through the gases on to a thermopile. The flue gas was analyzed with a standard Orsat apparatus and the temperatures recorded were measured with a thermocouple and a potentiometer. The loss of fuel through the grate was determined by weighing the refuse from the whole series of tests, and deducting the weight of ash, calculated from the weight of coal burned, and the known ash content of the coal.

It may be noted that the average smoke density in all of the tests was negligible when it is considered that a density of 23 per cent as recorded by the meter corresponds to a Ringelmann number of 1. The smoke density did not at any time reach a density greater than 15 per cent as measured by the recording meter when corrected for the zero reading of the instrument.

It may also be noted from the table, that, in spite of the fact that the fuel bed received no attention whatever except when each charge was placed, the furnace operated with a good overall efficiency at all of the different rates of burning. It is believed that the grate loss, which was quite high, can be greatly reduced by a slight change in the position of the horizontal rocking grate, as in its present position most of the ash is shaken through at the front edge, whereas it should be arranged so that the greater part of the ash is shaken through at the rear edge next to the rear wall of the furnace.

Fig. 4 shows a graphical log of the CO_2, the flue gas temperature, the temperatures of the air entering and leaving the casing, and the draft used, in a 12 hour test in which the position of the cross damper was occasionally changed as much as was necessary to maintain a constant heat output of approximately 35,000 Btu per hour from the casing. The coal used in this test was a 2 in. by 3 in. preparation from Franklin County, Ill. Since it was possible to maintain a continuous output equal to more than half of the rating of the furnace for a period of 12 hours, the test shown in Fig. 4 indicates that a furnace of this type would not need to be fired more often than twice a day except in unusually cold weather. Except for a short period following the placing of a charge of fresh coal, the heat output of the furnace was readily controlled by adjusting the position of the cross damper. The fact that all of the available draft was not being used when the heat output was low indicates that a thermostatic control of the damper could have maintained a uniform heat output throughout the cycle, except at the beginning.

A furnace of this type could be placed on thermostatic control immediately after placing a charge of fresh coal without fear of explosion or *puff-back.* It is probable that some over-heating at the beginning of the cycle might be

experienced because of the high combustion rate developed during the few minutes that the firing door was open in the process of preparing the fuel bed and placing the coal. However, it does not appear that the overheating would be sufficient to cause discomfort, and it is believed that the rapid response of the furnace after placing a charge is a very desirable performance characteristic.

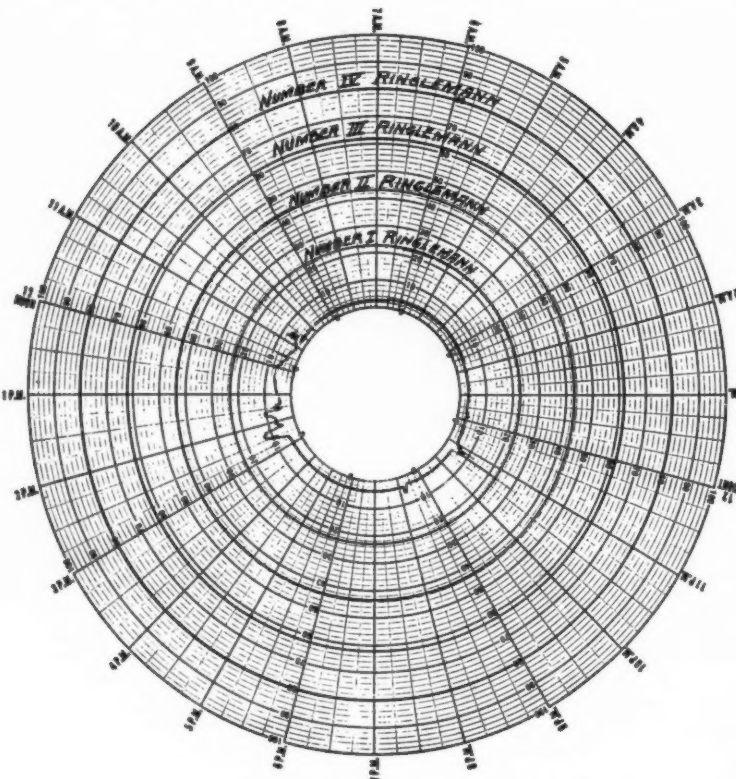

FIG. 5. REPRODUCTION OF AN ACTUAL SMOKE RECORD

The necessity for overcoming the phenomena of *puff-back* is principally responsible for the fact that the furnace did not maintain high CO_2 throughout the whole cycle. Sufficient primary coking air must be suppplied to carry away the gases released in the coking process during the time when that portion of the charge which is close to the firing door is being coked. The minimum amount of coking air required to eliminate *puff-back* determines the adjustment of the coking air damper, and it is impractical to attempt to extend the coking of a charge of fresh coal over the duration of a complete firing cycle. The requirements for secondary air are therefore considerably greater during the

first part of each cycle, and it is impossible with one setting of the secondary air damper both to provide a sufficient amount at that time and to avoid an excess during the later part. Tests have shown that, by careful adjustment of the three air dampers, CO_2 readings as high as 17 per cent can be obtained without smoke. Covering each charge with fine coal reduces the tendency toward *puff-back*. This makes it possible to reduce the amount of secondary air used, and effects a corresponding increase in the average CO_2. More frequent firing of smaller charges of coal would result in a higher CO_2 average.

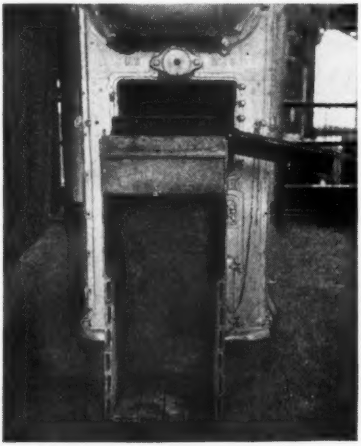

Fig. 6. Photograph of a Down-draft Conversion Burner. The Photograph was Taken with the Burner Standing on End in Front of the Furnace with the Bottom Side Facing the Camera

Reducing the amount of secondary air at the midpoint of each cycle would also have the same effect.

Fig. 5 is a reproduction of the actual smoke record made at the time the data shown in Fig. 4 were recorded. It also includes the record for the following 12 hour cycle, for which the same coal preparation was used. At the time this record was made, the recorder was adjusted to a zero reading of 4 per cent, so that it is only readings in excess of that amount that indicate other than a perfectly clear stack. It is possible to burn all types of high volatile bituminous coal in a down-draft coking furnace without producing any smoke whatever but any one of a number of unusual situations may cause a light smoke at times. The trace of smoke which appeared at times during the fore part of the test cycle was probably caused by the fact that the accidental positions of the various individual pieces of coal making up the charge were such that an uneven distribution of the gases passing under the baffle wall resulted. Thus, while sufficient secondary air entered the furnace

and was distributed uniformly across the lower face of the baffle, it is possible that there was a slight deficiency at one or more points. The use of a smaller sized coal reduces the chance of unequal mixing of the secondary air and coal gases, and increases the chance for obtaining 100 per cent smokeless combustion. It may be noted that no smoke was produced in the second cycle shown in Fig. 5 except for a few minutes immediately after firing. The small amount of smoke produced at the beginning of this cycle was caused by suddenly checking the fire subsequent to the very high combustion rate that was built up while the firing door was open and the fuel bed was being prepared and the charge placed. However, if the position of the Ringelmann number 1, obtained from a calibration of the recorder, is noted, it becomes evident that the amount of smoke produced in the 24 hours of operation was negligible.

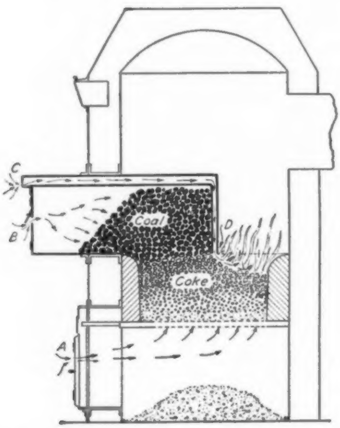

FIG. 7. VERTICAL CROSS-SECTION OF TEST FURNACE WITH RADIATOR REMOVED AND WITH DOWN-DRAFT BURNER IN PLACE

Allowing the fuel bed to burn too low before refueling, a situation apt to occur only in very mild weather, will of course make it impossible for the operator of a down-draft coking furnace to avoid the production of a light smoke for a more extended period. However, if the fore part of the inclined pin-hole grate is properly cleared of ash, a hot flame is soon established below the baffle wall and the smoke produced is reduced to a minimum.

DOWN-DRAFT CONVERSION BURNER

The down-draft coking principle was first applied to conventional updraft furnaces by means of the Azbe baffle developed by Victor J. Azbe, St. Louis, Mo., about 1926. The results of tests with the Azbe baffle made under Mr. Azbe's direction by Prof. P. E. Mohn at the Firing School in St. Louis during the summers of 1927 and 1928 suggested the idea of the down-draft conversion burner to the senior author in January 1936. As a result of a large number

of tests in more than a score of home furnaces and the construction of nearly sixty experimental models, it is believed that a practical design has been developed. Fig. 6 is a photograph of the burner used in numerous tests[3] made in the Mechanical Engineering Laboratory at the University of Illinois. Fig. 7 shows a longitudinal vertical section of a warm air furnace with a burner in place. Fig. 7 shows the condition of the fire immediately after placing a charge of fresh coal. The principle of operation is exactly the same as that of the down-draft coking furnace. Each charge of fresh coal is coked in the burner while the coke from the preceding charge is burned in the fire pot of the furnace. The secondary air enters the burner through a slot above the firing door, passes through a horizontal air passage in the hollow roof, and then through vertical passages in the rear and side walls. It then emerges at the bottom of the walls where it mixes with the gases from the coking coal. The gas-air mixture is ignited by coming in contact with the incandescent coke from the previous charge as in the down-draft coking furnace. A down-draft conversion burner can be installed in a conventional furnace in a few minutes without making any changes in the furnace other than the insertion of the burner through the firing door opening. After the installation of the burner a conventional furnace possesses all of the advantages of the down-draft coking furnace except for the absence of the inclined pin hole grate.

During the winter of 1941-42 a down-draft conversion burner was installed in a warm air furnace heating a two story wooden building having a calculated heat loss of 346,000 Btu per hour based on an assumed outdoor temperature of 0 deg F. A record was kept of the fuel consumed in the furnace and of the temperature maintained in the living quarters of the structure. Simultaneous fuel consumption and temperature records were also kept for an identical building heated by an identical furnace fired by the same fireman from the same pile of Jackson County, Illinois coal. Records were kept for a six weeks' period covering almost a complete range of winter weather conditions The results proved that less fuel was burned in the burner equipped furnace in every type of weather experienced during the test, and that the average saving in fuel effected by the burner was 24.1 per cent, based on the weight of fuel burned in the other furnace. The temperature in the living quarters was much more uniform in the building that was heated with the burner equipped furnace, and though there was far less overheating the average temperature exceeded by 3.8 degrees the average temperature maintained by the furnace that was used in the conventional manner.

OTHER POSSIBLE APPLICATIONS OF THE DOWN-DRAFT COKING PRINCIPLE

Since the down-draft coking principle applies only to the method of burning the fuel and is not affected by the design of the heating surface for utilizing the heat liberated, it is believed that it can be applied to stoves, boilers and water heaters as easily as to warm air furnaces. An attempt is now being made to apply the principle to a simple space heating stove. The authors believe that the down-draft coking principle offers the closest possible approach to 100 per cent smokeless combustion of high volatile fuels in hand-fired heating appliances, and an attempt will be made to develop all of the many possible applications in the near future.

[3] Burning Illinois Coal Smokelessly in Hand-Fired Heating Plants by J. R. Fellows and J. C. Miles. (Circular No. 43, Engineering Experiment Station, University of Illinois.)

APPENDIX

Kindling a Fire. The correct procedure to use in starting a fire in a cold furnace of the down-draft coking type is to clear the greater part of the inclined pin-hole grate, fill the firepot with coal to a level 3 to 4 in. below the bottom of the baffle, place several short sticks of dry wood on top of the coal and ignite the wood with a few pieces of crumpled paper. In 3 or 4 hours, after the starting charge is thoroughly coked, the coke in the fire pot may be broken up and a charge of fresh coal placed in the coking chamber. It is easier to start a fire in a down-draft coking furnace than in a conventional furnace because the fire pot is more accessible from the firing door opening, and while it is impossible to avoid the production of a light smoke from the starting charge, the coal is ignited at the top and burns from the top down with plenty of secondary air supplied directly over the surface and, as a result, the density of the smoke produced by the starting charge is much lower than the minimum density produced in kindling a fire in a conventional furnace.

DISCUSSION

W. M. MYLER, JR., Columbus, Ohio (WRITTEN): It is very gratifying to know that the Experiment Station of the University of Illinois has been able to successfully apply the down-draft coking principle to a hand-fired domestic furnace which is simple enough in operation to have a reasonable chance of performing satisfactorily in the hands of the average housewife.

It is particularly gratifying that this system of combustion can be accomplished satisfactorily by means of a conversion burner which can be installed in an average furnace. If this were not possible, the use of this method of combustion would be postponed indefinitely.

In normal peace times, the smokeless feature of the operation would undoubtedly be the one which would command primary interest. At the present time, when transportation facilities are strained to the utmost in supplying fuel for all purposes, anything that would substantially increase the fuel economy of a domestic hand-fired furnace should be given whole-hearted endorsement. It is, of course, too much to hope that in the hands of the average housewife as high efficiency would be obtained as that reported in these tests. If we consider, however, that the average housewife with her present equipment is attaining probably only a maximum of 50 per cent overall efficiency, and that with this new equipment it appears to be possible for her to obtain half the possible increase, thus permitting her to obtain 60 per cent efficiency, this would represent a fuel saving of approximately 16⅔ per cent, which is certainly worthwhile. In addition the smokeless benefits would be achieved at the same time.

It would seem that with the prospects of such a large increase in efficiency within the range of probability, it might be possible to obtain from the government powers having jurisdiction an allocation of sufficient metal to make up conversion burners, at least in limited quantities, to take advantage of this fuel saving.

In reviewing the data presented in Table 1, the question has arisen regarding the item *grate loss, Btu per pound of fuel,* which is shown as 847 Btu per pound of fuel for all rates of combustion. In view of the fact that the weight of ash was determined as a total for the entire series of tests, this is the only way that this could be recorded. It has been the writer's past experience, however, that the grate loss is not always the same for all rates of combustion. Quite frequently at the lower rates of combustion, more unburned carbon will find its way into the ash and consequently

the reported results may show too high an efficiency for the lower rates of combustion and probably not quite sufficiently high efficiency for the two higher rates of combustion.

The effect of such a change would be, of course, to flatten out the efficiency curve and actually give an increase in efficiency at the higher rates which are the rates at which the furnace would be operating during a considerable portion of the time and perhaps at which it would show a larger saving of fuel.

C. F. Newport, Chicago, Ill. (Written): A hand-fired boiler or furnace that will warm the home to a desired temperature for long periods without attention is most desirable. Long firing periods can be had by burning anthracite; and long coaling periods with even room temperatures under varying weather conditions can be had when burning the smaller sizes of anthracite in a magazine feed type boiler.

A long firing period with bituminous coal is more difficult to obtain. The authors seem to have achieved long firing periods when burning high volatile bituminous coal by coking the coal before burning it and thus eliminating objectionable smoke. Instead of feeding the coal to the fire by gravity from a storage magazine (which is a difficult proposition with soft coal) they feed the coal to the fire by hand after it has been coked in a coking chamber that holds enough coal for one firing period or cycle.

The draft for the furnace is held constant (apparently by a barometric draft control) and the heat output is controlled by a choke damper close to the furnace. It seems doubtful if this damper can be controlled by an ordinary electric thermostat and damper motor because of its off and on positions, but the ideal automatic control would be one obtaining a gradual control by varying the position of the damper to suit the room temperature requirements.

Will gases escape through the air inlet to the coking chamber or through the air inlet to the baffle, if the choke damper is closed enough to hold the fire in check, when little or practically no heat output is required?

Since air entering the furnace through the three air inlet ports must be proportioned correctly for proper results it seems that the correct setting of each of these ports will require the services of an expert. Even though they were set at the factory it would still be necessary to adjust the barometric control in the smoke pipe to maintain a constant draft pull through the furnace. These conditions can be obtained in a laboratory, but for successful operation similar conditions must be obtained in the field, where varying conditions of installation prevail.

Will tar and condensed moisture form on the walls and door inside the coking chamber, when burning certain types of coal?

As the method of firing is different from that with which the ordinary home owner has become familiar he will very likely require special instructions on how to fire for proper results. Although the performance of the furnace has shown such good results from the tests, it remains to be seen how it will perform in the field.

R. A. Sherman, Columbus, Ohio (Written): The commendation which it is customary for a discussor to extend in his opening remarks has been well earned by the authors as the senior author has spent seven years to develop the furnace described. However, in consideration of the difficulties of the task of accomplishing smokeless combustion in a non-mechanical device and the many man-years of effort that have previously been expended on the task, this price is not great.

The opening statement of the paper that the temperature required to ignite a coal gas-air mixture is much higher than that required to release the volatile matter from the coal is somewhat exaggerated. Although it is true that volatile starts to be driven from coal at 700 to 900 F, the range in which it is rapidly released is in the same range as the ignition temperature of the gases, that is, 1100-1200 F.

The difficulty from smoke in hand-fired furnaces is usually not that a high enough temperature does not exist at some place in the furnace for the ignition of the

volatile, but that this temperature is not maintained in the zone where the volatile can meet air for its combustion.

Control of the rate of liberation of the volatile matter to match the rate of admission of secondary air is essential for smokeless combustion. The authors control that rate by the rate of admission of coking air through the charge of coal. It is interesting that they have found it possible to place the control of the rate in the size of the orifice through which the air is admitted and independent of the resistance of the fuel bed. Data on the pressures in the various parts of the furnace during a complete cycle would be a valuable addition to the paper.

The efficiencies reported in Table 1, and in Fig. 3, are suprisingly high, and it is to be regretted that the heat balance data were not more completely given. The calculation of the ash pit loss as uniform for all rates is not justified but probably introduces no great error. No data are given for the loss in CO and it is assumed that they are included in the stack losses or that there were none.

The heat output and efficiencies given are obtained by difference between the heat input and the sum of the stack and ash pit losses. They thus include the radiation and any unmeasured losses. The values would be more convincing if at least one directly measured efficiency were given or if a test had been run with coke or anthracite with which the unaccounted losses are all very low.

To gain some conception of the agreement between the directly and indirectly measured efficiencies, the writer calculated the output in the high rate test reported in Table 1. The output was calculated from the value of 700 cfm, given as the approximate rate of air circulation, and the rise in temperature. This gave a value of 55,188 Btu per hour for the output and 57.1 per cent for the efficiency as compared to the values of 69,172 and 71.6 per cent which the authors calculated by difference. It is possible, of course, that the air flow was in error by that amount.

It is to be hoped that the authors will obtain further confirming data on the directly-measured output and efficiency in their future tests. High efficiency is a desirable feature, although possibly less essential than smokeless operation, but if the latter is obtained by complete combustion of the volatile hydrocarbons normally lost, high efficiency will naturally follow.

The furnace is not foolproof for, as the authors point out, it is essential that it be fired at proper intervals, but this is surely not too much to ask of the users if they are to receive the benefits described.

M. W. McRae, Chicago, Ill.: Did the authors have any difficulty with the stationary grate burning out due to most of the ash being deposited on the shaking grate, and also was any trouble experienced with breakage of the down-draft ducts, which it is understood were made of refractory?

Author's Closure: It is not clear whether the questioner about smoke backing out of the door had in mind the firing door or the secondary air ports. The authors have not experienced the slightest tendency toward smoke backing out through either of those openings. There is a very strong draft effect in those small vertical flues, in which the burning takes place. It has been the authors' experience that the furnace in the laboratory will operate successfully on a draft at the smoke collar as low as one-thousandth of an inch of water. As a matter of fact, the authors have actually operated, just as a stunt, with a slight positive pressure at the smoke collar. That can only be made possible by the phenomenon known as static pressure regain. As the gases pass through the radiator toward the smoke collar, the temperature of the gases decreases, the specific volume correspondingly decreases, the velocity decreases, and there is a chance for conversion of kinetic energy to static pressure. Operation has been obtained for short periods with positive pressures as high as five thousandths of an inch of water at the smoke collar. The authors can say definitely that they do not believe there will be any difficulty from that source.

Regarding the question of whether soot and tar gathered in the coking chamber: there is considerable accumulation of soot, but the authors have not noticed any

accumulation of liquid tar on the side of the coking chamber. There seems to be no more accumulation of soot on the inside of the firing door than in the conventional furnace.

In reference to special firing instructions being required, the technique of firing the furnace is extremely simple. It is important, of course, that the proper technique be followed. The technique consists simply of shaking the grate and pushing the coke back. The only rule is; just push the coke back as far as it can go. It pushes back very easily because of the shape of the floor of the coking chamber and the inclined pinhole grate. The authors have talked with a number of furnace salesmen and they seem to feel that it will not be difficult to instruct the public. In the first place, the operating instructions will be very simple; and in the second place, it is something different. The public will not expect to know how to operate it without instructions and therefore will probably follow the simple instructions that will be furnished.

It will be necessary, as in any hand-fired furnace, for the operator to use judgment in firing the furnace in very mild weather during the early fall and late spring. He will experience overheating if he puts in a charge of coal that would be suitable for cold weather. The proper procedure to use in mild weather is to fire a small charge of coal once every 24 hours, preferably in the evening, because invariably the nights are cooler than the days. The volatile matter will then be burned off through the night when there is more demand for heat and the coke will hold fire through the day without the production of much heat.

The authors thoroughly agree on the variation of grate loss with different combustion rates but as there was not sufficient time available for running long tests at each combustion rate they had no other way of determining it.

The authors disagree with Mr. Sherman's comment, that the opening statement of the paper is exaggerated. S. W. Parr and H. L. Olin reached the following conclusion after exhaustive studies in the coking of Illinois coal:[4] Coals of the Illinois type can be coked at a temperature approximately 400 deg to 450 C (450 deg C corresponds to 842 deg F). J. W. McDavid[5] concluded that a temperature of 1500 F is required to insure the instantaneous ignition of a coal-gas-air mixture when it is heated by the sudden application of a hot body.

Parr and Olin's experiments indicate that all of the smoke-forming hydrocarbons in Illinois coal are converted to the gaseous state at temperatures below 842 F, whereas McDavid's experiments indicate that a temperature of 1500 F is required to insure the ignition of all of them.

If smokeless combustion is to be achieved, all smoke-forming gases must be mixed with air and the mixture must be forced to pass some point in the furnace where the temperature is high enough to insure its ignition.

The authors agree with Mr. Sherman that further tests to determine the bonnet efficiency should be made at the first opportunity. While it is essential that the furnace be re-fired after the completion of the coking process and before the greater portion of the coke is consumed, this interval during which the furnace may be properly attended would continue for several hours under the operating conditions experienced in normal winter weather.

Concerning air distribution and control the authors refer those interested to a paper[6] recently published, in which it is pointed out that orifice size rather than fuel bed resistance determines the air distribution.

Mr. McRae asked about burning out of the stationary grate. This grate was carefully scraped clean and examined, just before leaving Urbana, after approximately 5 months of operation, part of the time at maximum combustion rate. Not the

[4] The Coking of Coal at Low Temperatures, University of Illinois Engineering Experiment Station *Bulletin* No. 60.

[5] The Temperature of Ignition of Gaseous Mixtures, *Transactions*, Chemical Society (London), Vol. 3, 1917, pp. 1003-1015.

[6] An Improved Hand-fired Furnace for High-volatile Coals, A.S.M.E. *Transactions*, April 1942.

slightest tendency toward distortion or oxidation was detected. Of course the burning coke does lie directly on the grate and the grate is not protected by ash; however, some air is continually passing through the grate and the cooling seems to be sufficient to prevent overheating. The pinhole construction prevents localized areas of intensive burning and this is a help in preventing damage to the grate. This arrangement also reduces the tendency toward formation of clinker. As a matter of fact, it definitely eliminates the formation of clinker. The authors have not had any tendency toward clinker formation with any types of coal used, and some of the coals used have been coals like catlin coal, which has a very strong tendency toward clinkering. In other words, it has an ash with a very low fusion temperature.

Mr. McRae also questioned the durability of the downdraft refractory flues. The authors have not had as much experience with that construction for the baffle wall as they would have liked, but no difficulty was experienced up to this time. The refractory problem was discussed at length with officials of a large refractory company. The furnace in its present stage is purely and simply a refractory problem, and its success depends on whether or not any difficulty with the refractory arises. The authors have been experimenting with refractory shapes in the construction of the baffle wall for about two years, and they have had no difficulty with any of the shapes used. Unfortunately, the authors have not had a long experience with the particular design that is being used at present.

1238

No. 1238

FINAL VALUES OF THE INTERACTION CONSTANT FOR MOIST AIR

Third progress report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Towne Scientific School, University of Pennsylvania.

By JOHN A. GOFF,* J. R. ANDERSEN,** AND S. GRATCH,† PHILADELPHIA, PA.

INTRODUCTION

THIS paper reports final values of the interaction constant for moist air for the range 5 to 25 C as determined experimentally in a cooperative investigation between the Towne Scientific School, University of Pennsylvania, and the American Society of Heating and Ventilating Engineers through its Research Technical Advisory Committee on Psychrometry. The ultimate objective of the investigation is a formulation of the thermodynamic properties of moist air which, on account of accuracy and thermodynamic consistency, can claim universal acceptance as standard. To accomplish this objective requires first of all to measure a certain temperature function, called the interaction constant, reliable values of which have hitherto been unknown.

A preliminary value of the interaction constant for 15 C was reported in a previous paper.[1] This preliminary value was much smaller than had been anticipated in designing the apparatus for its measurement so that the allowed errors in individual measurements did not yield a sufficiently small probable error in the final result. It became clear, therefore, that the apparatus itself and the technique of operating it required further study with a view toward improving the reliability of individual measurements. Furthermore, it was necessary to make measurements at other temperatures in the range of interest to the air-conditioning engineer. The improved reliability and extension to cover the range 5 to 25 C have now been accomplished.

Although the interaction constant is defined in Chapter 1, Heating, Ventilating, Air-Conditioning Guide 1943, and although the theory of the experiment devised for its measurement has been explained in the Preliminary Report referred to, it seems desirable to review these matters briefly here.

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[1] A.S.H.V.E. Research Report No. 1186—The Interaction Constant for Moist Air by John A. Goff and A. C. Bates. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., June, 1943.

THE INTERACTION CONSTANT

Statistical mechanics predict that for a mixture of two individual gases, the relation between pressure P, specific volume v, and temperature T, valid at sufficiently low pressures, is

$$Pv = RT - [A_{aa}x^2 + 2A_{aw}x(1-x) + A_{ww}(1-x)^2]P \quad \ldots\ldots (1)$$

where the subscripts have been chosen with reference to moist air as a mixture of dry air (subscript a) and water vapor (subscript w) in mind. In this case x would denote the mol-fraction of the dry air. In the special case $x = 0$, the mixture consists of water vapor alone and only the term A_{ww} remains as coefficient of the pressure in the righthand member. This term is called the *second virial coefficient* of water vapor. It expresses the effect of intermolecular forces between pairs of like water molecules and is supposed to depend only on the temperature. Similarly A_{aa} denotes the second virial coefficient of dry air. Reliable values of A_{aa} and A_{ww} are available in the literature.

Complete and accurate information regarding dry air and water vapor separately cannot be made to yield complete and accurate information regarding mixtures of these two constituents; for superimposed on the forces between pairs of *like* molecules are the forces of interaction between pairs of *unlike* molecules. The effect of these interaction forces on the relation between pressure, specific volume, and temperature in the case of moist air is given by the coefficient A_{aw} called the *interaction constant.* It too is supposed to be a pure temperature function (not actually a constant) and its measurement is the immediate objective of the cooperative investigation being reported.

THE SATURATION-ISOTHERM EXPERIMENT

The experiment devised for the measurement of the interaction constant A_{aw} is referred to as a saturation-isotherm experiment. A steady stream of dry, carbon dioxide-free air is passed over liquid water in a suitable saturator and made to leave, *presumably saturated,* at measured pressure p and measured temperature T. It then pases through suitable dryers in which the water added in the saturator is removed and collected. Now the equation for saturation can be derived from Equation (1) by applying well-known identical relations of thermodynamics, except that certain information from the literature must be added as follows: (*a*) the vapor pressure of pure water, a function of temperature only, p_s; (*b*) Henry's constant expressing the extent to which air dissolves in liquid (or solid) water, regarded as a pure temperature function, k; (*c*) the specific volume of liquid (or solid) water regarded as depending only on temperature, v'_w. Details of the calculation are given in the Preliminary Report and only the final result will be repeated here, namely,

$$\log_e \left[\frac{(1-x)\,P}{p_s}\right] = A\left(\frac{P}{p_s} - 1\right) - B(1-\lambda)\left(\frac{P}{p_s} - 2 + \frac{p_s}{P}\right) \quad \ldots\ldots (2)$$

$$\text{where } A = -kp_s + \frac{v'_w\,p_s}{RT} + \frac{A_{ww}p_s}{RT}$$

$$B = \frac{A_{aa}p_s}{RT} + \frac{A_{ww}p_s}{RT}$$

$$\lambda = \frac{2A_{aw}}{A_{aa} + A_{ww}}$$

The parameter λ is obviously a pure temperature function if A_{aa}, A_{aw}, A_{ww} themselves depend only on temperature as predicted by statistical mechanics. The name *interaction constant* can be used alternatively for λ and A_{aw} without risk of serious confusion.

The mol-fraction of water vapor $(1-x)$ in the saturated mixture can be calculated if the weight of dry air passing through the apparatus during the same time that the water added in the saturator is later removed and collected in the dryer is determined. Then, since pressure P and temperature T are measured, there would remain only one unknown in Equation (1), namely, the interaction constant λ. But it was considered essential to avoid the necessity of weighing the dry air. Consequently it was arranged to return the dry air to a second saturator at lower pressure P_2, but same temperature T, and thence to a second dryer. The water added in the second saturator is collected over the same period of time as that in the first saturator; and since the same weight of dry air passes through both saturators during this time, this weight of dry air can be eliminated between two writings of Equation (2), one for the high pressure P_1 and the other for the low pressure P_2. This elimination makes possible the determination of λ without requiring to weigh the dry air; mathematically,

$$\lambda = \frac{\log_e \left(\frac{w_1 P_1}{w_2 P_2}\right)}{B(\pi_1 - \pi_2)\left(1 - \frac{1}{\pi_1 \pi_2}\right)} + 1 - \frac{A}{B\left(1 - \frac{1}{\pi_1\pi_2}\right)} - \Delta\lambda \quad \ldots\ldots\ldots \quad (3)$$

where

$$\Delta\lambda = \frac{\log_e \left(\dfrac{\left(1 + 1.60768 \frac{w_1}{w_a}\right)\left(1 - \frac{1}{\pi_1}\right)}{\left(1 + 1.60768 \frac{w_2}{w_a}\right)\left(1 - \frac{1}{\pi_2}\right)}\right)}{B(\pi_1 - \pi_2)\left(1 - \frac{1}{\pi_1\pi_2}\right)}$$

and where

w_1 = weight of water collected after saturation at the high pressure P_1
w_2 = weight of water collected after saturation at the low pressure P_2
$\pi_1 = P_1/p_s$
$\pi_2 = P_2/p_s$
w_a = weight of dry air from which w_1 and w_2 are removed simultaneously.
1.60768 = ratio of molecular weight of water to that of dry, CO_2- free air

The fact that w_a remains in Equation (3) would appear to contradict the statement that this has been eliminated between two writings of Equation (2). Actually, however, no contradiction is involved. For, since $\Delta\lambda$ is a small corrective term, a tentative value of λ can be computed from Equation (3) ignoring it; this tentative value can be substituted into either writing of Equation (2) to calculate an approximate value of $(1-x)$; and from it can be calculated an approximate value of w_a for the evaluation of $\Delta\lambda$ and the subsequent re-evaluation of λ itself.

EVIDENCE OF COMPLETE SATURATION

It is proposed to base calculations of the volume, enthalpy and entropy of moist air on the values of λ determined in the manner described. It is therefore important to adduce as much evidence as possible to show that these do not represent merely an empirical adjustment of Equation (2) to fit observed data. The most striking evidence on this point is presented under heading Interaction Constants in the form of a comparison between the measured values of λ and those predicted by quantum statistical mechanics. But the experimental part of the present investigation was practically completed before any such comparison was attempted; therefore, it seems appropriate to at first confine attention to the strictly experimental evidence.

In the first place, if λ is a pure temperature function as assumed in the derivation of Equation (2), then the same result should be obtained at a given temperature regardless of the pressure settings P_1 and P_2, regardless of the rate of flow through the apparatus for given pressure settings (within limits, of course), and regardless of whether saturation is approached by evaporation or condensation. Actually, the different elements of the apparatus were so critically matched that wide variations in these conditions would have introduced excessive uncertainties in the individual measurements and thereby have increased the probable error in the final result to the point of obscuring the evidence sought. Nevertheless, sufficiently wide variations to indicate the soundness of the method and the practical attainment of complete saturation were made.

In the second place, provision was made to weigh the dry air as a check on the satisfactory performance of the apparatus as a whole and as an additional check on the soundness of the method. It was explained in the preceding article how w_a could be eliminated between two writings of Equation (2), thus permitting the determination of λ without requiring to weigh the dry air. It was also explained that this value of λ could be substituted into either writing to calculate the *expected* weight of dry air passing through the apparatus during the time in which the two weights of water were collected in the dryers. The agreement between expected and directly measured weights of dry air was always so close that any slight discrepancy could be attributed entirely to uncertainty in the measurement of temperature ($\pm$ 0.02 C) or to uncertainty in the knowledge of the vapor pressure p_s, or both.

In view of the evidence briefly presented it seems safe to conclude that the complete saturation assumed in the derivation of Equation (2) was practically attained and that the measured values of λ are indeed those of the interaction constant contemplated in Equation (1).

REFINEMENTS

A tentative value of λ at 15 C, namely, 0.075 was given in the Preliminary Report; the final value reported now is substantially lower, namely, 0.048. This lowering is the direct result of various refinements of the apparatus and of the technique of operating it, which will now be discussed.

1. ***Pressure Control:*** The desirability of improving the pressure control was mentioned in the preliminary report. The source of air is a small oxygen bottle suspended from one arm of a sensitive balance. In order to limit the pressure in the

take-off tube and keep it reasonably constant, a small reducing valve was mounted directly on the cylinder. This valve was arranged for pneumatic control by connecting the space on the control side of its diaphragm to a suitable reservoir filled with compressed air. The pressure in the reservoir could, of course, be raised or lowered by admitting or withdrawing air.

During test, the pressure in the supply bottle would drop from about 1500 to 200 lb/in.,^{2} too wide a variation for a single valve to reduce to zero within a few thousandths of an inch of mercury as desired. Consequently a second reducing valve, mounted on the main control panel, was placed in series with the first one. But even the two valves in series permitted the pressure in the saturators to drift; hence some additional compensation was required. Shortly after obtaining the data reported in the Preliminary Report, one of the authors of that paper (A.C.B.) completed the construction of a device which provided the necessary additional compensation.

The device referred to consisted of a tall manometer connected to the delivery side of the first reducing valve and provided with a lamp and photocell on opposite sides of its mercury column at the desired level. If the mercury level should drop thus permitting light from the lamp to reach the photocell, this light would actuate the photocell to close a suitable relay and energize a resistance heater wound around the reservoir communicating with the control side of the reducing valve. The heat thus added would raise the air pressure in the reservoir, increase the opening in the reducing valve, and restore the mercury level in the manometer to its original position. By means of this device, the delivery pressure from the first reducing valve could be regulated to within ± 0.15 in. Hg thus permitting the second valve to regulate the pressure in the saturator to within a few thousandths of an inch of mercury over periods as long as 9 hours. Additional compensation in the form of a manually operated resistance heater wound around the air reservoir communicating with the control side of the second reducing valve was provided; but only rarely was it found desirable to use it.

2. *Stirring Motors:* The stirring motors were originally mounted directly over the thermostats to which the main manometers were also attached. This made it possible for vibrations to be transmitted to the manometers to produce small standing waves on the mercury menisci. This source of error was easily removed by building a separate scaffold on which to mount the stirring motors so that vibrations were damped out in the concrete floor of the laboratory.

3. *Rocking the Saturators:* Originally it was thought necessary to rock the saturators slowly in order to flush all inside surfaces with water and thus aid complete saturation. But it was immediately discovered that rocking produced small pressure fluctuations which introduced uncertainty into the readings of the manometers. Fortunately, preliminary tests had indicated that rocking the saturators was an unnecessary precaution; hence it was stopped with the saturators carefully levelled.

4. *Meniscus Corrections:* With the uncertainties described in the measurement of pressure eliminated or considerably reduced, it became feasible to achieve further refinement by applying more accurate meniscus corrections. These were obtained from data in International Critical Tables.

5. *Thermal Expansion of Mercury and Scales:* The temperature correction for relative thermal expansion between the mercury and the steel scales of the manometers was recalculated from data in International Critical Tables. The coefficient used was 0.9478×10^{-4} (F^{-1}). All pressures were reduced to inches of mercury at 68 F, 980.196 cm/sec^2 (Philadelphia). During the later tests, the temperature was measured at several points along the mercury column in order to obtain a more reliable average value on which to base the correction for thermal expansion.

6. *More Frequent Readings:* In order to further increase the accuracy of pressure measurements, readings were taken at more frequent intervals than in the preliminary tests. Intervals of ten minutes were used in the shorter runs, 15 min in the longer ones, so that at least twenty separate readings were obtained in any one run. The average of all readings was used in the final calculation of λ by means of Equation (3).

FURTHER REFINEMENTS

The refinements described in the preceding paragraph relate to the regulation and measurement of pressure. In this paragraph attention is turned to refinement in the collecting and weighing of the water removed by the dryers.

The drying agent used was anhydrone, $Mg\ (ClO_4)_2$, contained in glass towers. Approximately 50 g (grams) of anhydrone was introduced into each tower by means of an apparatus especially designed to minimize contact with air during the filling operation. It was desired to determine the increase in weight to within $\pm$ 0.0005 g representing an accuracy of about one part in 100,000. The chief sources of error appeared to be buoyancy, variable conditions of the outside surfaces of the towers, and other effects produced by changes in atmospheric conditions.

1. *External Buoyancy:* This source of error was effectively eliminated by using as counterweight a sealed dummy tower similar in all respects to the main towers. This was possible since change in weight between beginning and end of test was the only thing of interest.

2. *Internal Buoyancy:* Due to the nature of the drying chemical and to the method of filling the towers, there was a considerable volume of air inside the towers. It was obviously important to account for any possible changes in the weight of this air between beginning and end of test. During weighing, a tower was necessarily open to atmosphere through a capillary tube; hence any change in atmospheric pressure between beginning and end of test (possibly 9 hours) would produce a change in the weight of air within the tower. This effect was over-looked in the preliminary tests but was compensated for in the final tests by unsealing the dummy tower for a short time before each weighing operation thus permitting it to make the same adjustment to atmospheric conditions as the main towers.

An additional internal buoyancy effect was produced by displacement of air from the towers by the water collected. This effect is difficult to determine accurately because the water is absorbed as water of crystallization and the exact hydrate formed is not definitely known. Assuming that $Mg\ (ClO_4)_2 \cdot 7\,H_2O$ is the hydrate formed, the buoyancy correction calculated from density data given in International Critical Tables was 0.0008 g per gram of water collected. Information regarding the densities of other possible hydrates is not very reliable; but the evidence is that the correction would not be materially altered.

3. *Surface Effects:* Deposition of moisture on the outside surface of a tower will produce an error in weighing unless precautions are taken to insure that similar deposition occurs on the dummy tower also. The various precautions include storing all towers in a dessicator when not being used or weighed, giving each tower a standard wiping with a clean rag before placing it in the balance case, keeping a dish of concentrated sulfuric acid inside the balance case, and allowing ample time (25 min) before each weighing for the new tower to come to equilibrium with the dummy tower inside the balance case. The tediousness of the weighing operation can be appreciated from these remarks.

4. *Other Effects:* Rapid temperature changes and temperature gradients in the neighborhood of the balance produce some convection currents inside the balance case. To thermostat the laboratory room was impracticable, but it was found possible to minimize these effects by periodically adjusting the opening of the door to the warmer laboratory adjoining.

Data from International Critical Tables indicate that approximately 4×10^{-4} milligrams of water per gram of air passes through the anhydrone. An attempt was made to obviate the necessity of applying this correction by using a P_2O_5 backing tower. But the weight of water collected in the backing tower turned out to be too small to measure accurately; hence the attempt was abandoned and the correction applied instead.

OPTIMUM OPERATING CONDITIONS

As a guide to the selection of optimum operating conditions, an approximate relation giving the error in λ produced by errors in the individual measurements was derived from Equation (3) as follows:

$$d\lambda = \frac{1}{B(\pi_1 - \pi_2)}\left[\frac{1.6(\pi_1 - 1)}{w_a} dw_1 + \frac{1.6(\pi_2 - 1)}{w_a} dw_2 + \frac{d\pi_1}{\pi_1 - 1} + \frac{d\pi_2}{\pi_2 - 1}\right] \quad . \; . \; . \; (4)$$

where $d\lambda$, dw_1, etc, are respectively the *absolute* errors in λ, w_1, *etc.*

In all final tests P_2 was set at approximately 25 in. Hg, just enough below atmospheric pressure to insure that the stopcocks on the drying towers remained tight. A lower setting than necessary would not only invite leakage of air into the low pressure side of the system but would make small errors in the measurement of P_2 more significant in the evaluation of λ.

Having decided on a fixed value of P_2 it would appear from Equation (4) to be desirable to set P_1 as high as possible. However, with the existing apparatus it was not practicable to exceed about 80 in. Hg; hence a setting of approximately 75 in. Hg for P_1 was used in all final tests.

With the two pressure settings fixed, Equation (4) shows that the contributions to $d\lambda$ from small errors in w_1 and w_2 are minimized by selecting the largest possible value of w_a. In earlier tests, an upper limit on the value of w_a was set by the capacity of the supply cylinder from which the air was drawn. However, after having established close agreement between the expected and directly measured weights of dry air in a sufficient number of tests, this cross-check was discontinued and greater quantities of air drawn from the larger bottle were used. The capacity of the dryers then fixed the upper limit on w_a.

ESTIMATED UNCERTAINTY IN λ

Weighings were considered accurate to within ± 0.0005 g. Each tower was weighed twice in each test. The possibility that the buoyancy, surface, and other effects discussed under heading Further Refinements were not perfectly compensated introduces some additional uncertainty into the measurement of water collected; however, it seems fair to assign to w_1 and w_2 an over-all uncertainty of ± 0.0015 g per tower plus 10^{-4} milligrams per gram of air.

Uncertainty in the measurement of pressure arises from uncertainties in (*a*) reading the scales (± 0.001 in.Hg), (*b*) correcting for thermal expansion ($\pm 0.0004\ P$ in.Hg); (*c*) adjusting the scales (± 0.002 in.Hg for earlier tests, ± 0.001 in.Hg for later tests).

The estimated uncertainty of ± 0.02 C in the measurement of temperature contributes practically no uncertainty to the determination of λ even though it does correspond to an uncertainty of as much as ± 0.14 per cent in the saturation pressure, p_s. If this figure be increased to ± 0.20 per cent to allow for the possibility that the values of p_s from the literature may be in error by as much as ± 0.06 per cent, still the uncertainty in λ from both sources is at most ± 0.0005. This emphasizes an important advantage of the two-stage experiment over a single-stage experiment requiring to weigh the dry air. In the latter, small uncertainties in the measurement of temperature and in the

knowledge of vapor pressure would greatly reduce the accuracy of the calculated value of λ.

The uncertainty in the gas constant R is entirely negligible. That in the apparent molecular weight of dry, CO_2-free air may be estimated as ±0.01 per cent; but it contributes no appreciable uncertainty to the calculation of λ by the method of eliminating the weight of dry air. It is impossible to estimate closely the uncertainties in A_{aa}, A_{ww}, v'_w and k, all of which enter the calculation of λ. But assuming that the first three are in error by at most ±0.5 per cent and making a liberal allowance of ±10 per cent in k, the total contribution to uncertainty in λ is at most ±0.0003.

In allowing an uncertainty of as much as ±10 per cent in Henry's constant k, it was not intended to asperse available solubility data from which this constant is derived. Rather, this liberal allowance was made in recognition of the fact that k entered Equation (2) on certain simplifying assumptions. Similarly, the use of Dalton's Rule to eliminate mol-fraction x from the righthand member of Equation (2) represents an approximation which can be shown, however, not to contribute appreciable uncertainty to the calculated value of λ.

SUMMARY OF RESULTS

The final values of λ are given in Table 1. The estimated uncertainties given in Column 4 are the sums of the separate contributions explained in the preceding article. In Column 5 are given values of the probable error of the mean (Column 2) as computed by the conventional method except that an allowance of ±0.001 for systematic errors in thermal data from the literature has been included. In computing the mean value of λ for a given temperature the values from individual tests were assigned different weights to favor those showing greatest internal consistency. The exact method of weighting need not be explained since if all tests had been assigned unit weight the final results would not have been altered materially.

The method of reducing the data of Pollitzer and Strebel [2] to obtain values of λ for comparison with those of the present investigation was explained in the Preliminary Report. It was not considered worthwhile to attempt to improve this method or to estimate the limits of uncertainty in the results. It may be, therefore, that the values listed in Table 1 for 50 and 70 C do not

TABLE 1—SUMMARY OF RESULTS

1 t (c)	2 λ (MEAN)	3 NO. OF TESTS USED	4 ESTIMATED UNCERTAINTY	5 PROBABLE ERROR OF MEAN	6 AUTHORITY
5	0.043	4	±0.003	±0.002	Goff, Andersen, Gratch
15	0.048	6	±0.003	±0.002	Goff, Andersen, Gratch
20	0.054	4	±0.005	±0.002	Goff, Andersen, Gratch
25	0.053	5	±0.005	±0.003	Goff, Andersen, Gratch
50	0.056	..	....	....	Pollitzer & Strebel
70	0.07	..	....	....	Pollitzer & Strebel

[2] Über den Einfluss indifferenter Gase auf die Sättigunge-Dampfkonzentration von Flussigkeiten, by F. Pollitzer and E. Strebel. (*Zeit. für Phys. Chemie*, Vol. 110, 1924, pp. 768-785.)

represent the best possible reduction of the Pollitzer and Strebel data; but they do afford valuable comparison.

PREDICTIONS FROM STATISTICAL MECHANICS

Statistical mechanics asserts that the second virial coefficient of a gas arises from forces between *pairs* of molecules, and that, in the aggregate, the force between any pair may be regarded as depending only on the distance of separation r. The existence of a force of this kind implies that the pair possesses potential energy $E(r)$. Statistical mechanics then proves that the following relation between the second virial coefficient $A(T)$ and the potential energy $E(r)$ is a good approximation to the truth in some cases.

$$A(T) = 2\pi N_0 \int_0^\infty (e^{\frac{-E(r)}{KT}} - 1) r^2 dr \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (5)$$

where N_o = number of molecules per mol of gas (Avogadro's Number)
k = Boltzmann's Constant
r = distance of separation

LONG-RANGE ATTRACTIVE FORCES

Recent developments in quantum mechanics have made it possible to calculate the function $E(r)$, at any rate, the contribution to $E(r)$ arising from long-range attractive forces. In an interesting article entitled, *Van der Waals Forces,* Margenau [3] reviews the present state of these developments. He states that all through the period from the beginning to the end of the nineteenth century, investigators were convinced of the novelty and essential uniqueness of these long-range attractive forces, but that now these forces have been stripped of their uniqueness and shown to be of the nature of simple electric interactions.

Some molecules, notably the water molecule, possess what is called a *permanent dipole moment* due to a permanent asymmetric distribution of the electric charges of their constituent atoms. Averaged over all possible orientations, the force between two such rotating molecules would be exactly zero; hence, if no particular orientation were favored, there could be no resultant force. But, as a matter of fact, attractive orientations *are* favored because they minimize the potential energy of the pair for a given separation r and because, in the aggregate, the number of pairs having a given potential energy is larger the smaller this energy. This so-called *dipole alignment effect* contributes to $E(r)$ an amount proportional to $p^2_1 p_2^2/r^6$ and inversely proportional to absolute temperature T. The dipole moments of the two molecules are denoted by p_1 and p_2, different subscripts being used to include the case of a pair of *unlike* molecules.

In the discussion of the dipole alignment effect no mention was made of the possibility that the distribution of electric charges within each member of a pair of molecules is actually deformed to some extent by the electric field surrounding the other molecule. This is so-called *induction effect* and it makes an additional contribution to $E(r)$ of amount proportional to $(a_1 p_2^2 + a_2 p_1^2)\ /r^6$,

[3] Van der Waals Forces, by H. Margenau. (*Rev. Mod. Phys.*, Vol. 11, No. 1, January 1939, pp. 1-35.)

where the a's denote the polarizabilities of the two molecules, different subscripts being used again to cover the case of unlike molecules. It should be noted that the induction effect is not strongly temperature dependent as is the case with the alignment effect.

In analyzing the dipole alignment and the induction effects, the atoms were regarded as structureless points carrying electric charges. But a closer examination reveals the necessity of considering the fact that each atom consists of a nucleus surrounded by a deformable electron cloud. This means that each atom may become, if only momentarily, a dipole even though it has no permanent dipole moment. The possibility that each atom, and therefore each molecule composed of atoms, may have a temporary dipole moment generates an additional tendency toward alignment and produces, in an aggregate of molecules, the so-called *dispersion effect*. The contribution to $E(r)$ from these dispersion forces is, in the main, proportional to $f_1 f_2 / r^6$ where the f's are constants characteristic of the interacting molecules and are obtainable from information regarding their indices of refraction.

Margenau[4] points out that the distinction between the three effects described is really historical and that fundamentally all three spring from the same root. Each may contain terms in higher powers of the inverse distance, namely, r^{-8}, r^{-10}, etc. But, as an adequate approximation, the contribution to $E(r)$ from *all* long-range attractive forces may be written

$$E_{-}(r) = -\left(\frac{a_1}{T} + a_2 + a_3\right) r^{-6} - a_4 r^{-8} - a_5 r^{-10} \quad . \; . \; . \; . \; . \; . \; . \quad (6)$$

Table 2 lists numerical values of the a's for gases of interest in the present paper. These values are computed from spectroscopic and other related information.

SHORT-RANGE REPULSIVE FORCES

It is well known that, for very small distances of separation r, the forces between molecules become strongly repulsive. The potential energy $E(r)$ must

TABLE 2—VAN DER WAALS FORCE CONSTANTS

	Gas	Dipole Alignment $a_1 \times 10^{60}$ (erg cm^6 K)	Induction $a_2 \times 10^{60}$ (erg cm^6)	Dispersion			Reference
				$a_3 \times 10^{60}$ (erg cm^6)	$a_4 \times 10^{76}$ (erg cm^8)	$a_5 \times 10^{92}$ (erg cm^{10})	
1	H_2	0	0	11.4	31	45	Margenau
2	N_2	0	0	57.2	120	130	
3	O_2	0	0	39.8	96	120	
4	H_2O	55700	10	47	114.9	127.9	
5	Dry Air	0	0	53.6	114.9	127.9	Authors
6	H_2-N_2	0	0	25.6	61	76.5	
7	Moist Air	0	5	50.2	114.9	127.9	

[4] Loc. Cit. See Note 3.

therefore rise rapidly with decreasing r. This means that there must be added to $E(r)$ as given by Equation (6) a positive term $E_+(r)$ representing the contribution from short-range repulsive forces. It appears to be possible to predict the form of this contribution from quantum mechanical considerations; but the prediction has not yet been worked out for any of the gases of interest here.

A practical way around the difficulty is suggested in Fig. 1. The true values of $E(r)$ are represented by the solid curve; but these can be replaced by the dotted curve which follows Equation (6) for values of $r > r_0$, but rises vertically to infinity at $r = r_0$. If the value or r_0 is adjusted to make $A(T)$

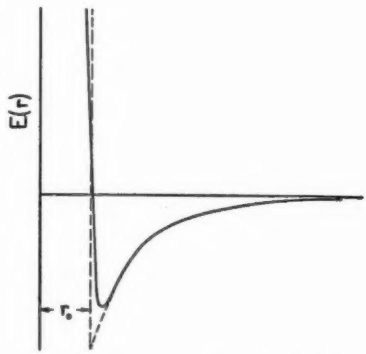

FIG. 1. POTENTIAL ENERGY $E\ (r)$

calculated from Equation (5) agree with the observed value at a temperature near the middle of the range of interest, this agreement can reasonably be expected to hold for other temperatures not too far removed from the one in question.

SECOND VIRIAL COEFFICIENTS

Details of the calculations will not be given in this paper, but the order of agreement between calculated and observed second virial coefficients for H_2, N_2 and dry air will be shown.

In the cases of H_2 and N_2, reliable information is available only at 0 C (Cragoe).[5] At other temperatures there is disagreement of as much as 20 per cent between various investigators. The only comprehensive survey of existing data is that of Keyes[6] whose correlation, however, is unsatisfactory from the point of view of the theory outlined here. Since a discussion of the second virial coefficients of H_2 and N_2 is incidental to a comparison between observed and calculated values of the interaction constant for H_2-N_2 mixtures,

[5] Slopes of the pv Isotherms of Some Thermometric Gases at Pressures Below Two Atmospheres, by C. S. Cragoe. (Temperature Symposium, *American Institute of Physics*, pp. 89-126.)
[6] Gas Thermometer Scale Corrections Based on an Objective Correlation of Available Data for Hydrogen, Helium and Nitrogen, by F. G. Keyes. (Temperature Symposium, *American Institute of Physics*, pp. 45-88.)

TABLE 3—SECOND VIRIAL COEFFICIENTS A (T) (CC/MOL)

GAS	0 C		20 C		$r_0 \times 10^8$ CM
	Observed	Calculated	Observed	Calculated	
H_2	−14.2	−13.9	−14.9	−15.4	2.975
N_2	+11.1	+11.1	+6.1	+6.16	3.457

and since the only observations available for this purpose are those of Verschoyle,[7] it seems that to use his data exclusively will at least be a consistent procedure. These data may be in error by as much as 10 per cent, though probably less. Table 3 gives the desired comparison.

In the case of dry air, it would be necessary to calculate the second virial coefficients for all individual constituents plus the interaction constants for all unlike pairs to obtain the second virial coefficient of the mixture, if strict adherence to the theory were to be attempted. However, since the principal constituents are N_2 and O_2 and since the various coefficients in Table 2 for these gases do not differ widely, it becomes unnecessary to follow this elaborate procedure. Instead the coefficients for the mixture can be obtained as weighted averages of the corresponding coefficients for N_2 and O_2. The results are given in Row 5 of Table 2.

Fortunately, the observed data on the second virial coefficient of dry air are quite satisfactory. Fig. 2 shows the order of agreement obtained by adjusting $r_0 = 3.396 \times 10^{-8}$ cm to give close agreement at 20 C. It is to be noticed that a constant value of r_0 gives only fair agreement over the range −10 to 80 C. However, this particular range is an unfavorable one, because in it the second virial coefficient changes sign; and when this occurs, the calculations are very sensitive to changes in r_0. Allowing r_0 to vary by as little as 1 per cent over this range would achieve perfect agreement with the Beattie-Bridgeman[8] data.

INTERACTION CONSTANTS

In order to obtain appropriate coefficients in Equation (6) for mixtures of two gases it is necessary to employ certain combination rules suggested by the theory though, in principle, these can be computed directly from the quan-

TABLE 4—INTERACTION CONSTANT FOR H_2–N_2 MIXTURES

	0 C		20 C	
	Observed	Calculated	Observed	Calculated
A_{H2-N2} (cc/mol)	−12	−8	−14	−11
λ	7.7	5.7	3.2	2.4

[7] Verschoyle, *Proc. Roy. Soc.* A, Vol. 111, 1926, p. 552.
[8] A New Equation of State for Fluids, by J. A. Beattie and O. C. Bridgeman. (*Proceedings American Academy Arts and Science*, Vol. 63, 1928, p. 229.)

tum mechanical equations. Thus, the a_1 for a mixture should be the *geometric* mean of the a_1's of the individual gases, since this coefficient is proportional to the product $p_1^2 p_2^2$. An *arithmetic* mean can be used for the coefficient a_2 if the polarizabilities of the two gases do not differ too greatly. The problem is more difficult in the case of the dispersion coefficients, though for a_3 it is clear that the *geometric* mean is the best approximation. Regarding the higher coefficients a_4 and a_5 *geometric* averaging is recommended especially since the corresponding terms in Equation (6) contribute relatively little to the final result.

Row 6 gives the coefficients for H_2-N_2 mixtures as computed by means of the combination rules explained above. Again using the Verschoyle data the interesting comparison shown in Table 4 is obtained.

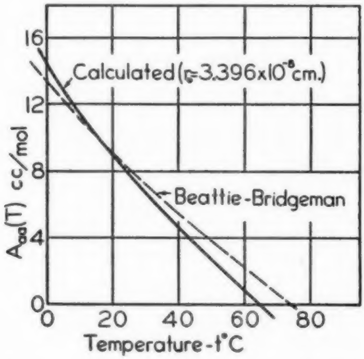

FIG. 2. SECOND VIRIAL COEFFICIENT FOR DRY AIR

As mentioned previously the Verschoyle data are not very accurate. In fact, judging from the values given by Cragoe[9] and by Keyes[10] for 0 C, they may be in error by as much as 10 per cent. Nevertheless, the comparisons of Tables 3 and 4 attest to the powerfulness of present theory, especially when it is noticed that, in the case of H_2-N_2 mixtures, λ is so much larger and depends so much more markedly on temperature than in the case of moist air.

Before attempting to calculate the interaction constant for moist air, some explanation of how the coefficients in Table 2 (Row 4) for water vapor alone were obtained, is in order. The values of a_1, a_2, a_3 were taken from Margenau[11] who in turn quoted London.[12] The higher coefficients a_4, a_5 were not given. However, since the ratios a_3:a_4:a_5 do not vary greatly amongst the various gases and since the values of a_3 for water and for dry air (Row 5) are not radically different, it was assumed that the same values of a_4 and a_5

[9] Loc. Cit. See Note 5.
[10] Loc. Cit. See Note 6.
[11] Loc. Cit. See Note 3.
[12] F. London, *Transactions Faraday Society* 33[3], 8, 1937.

could be used for water as for dry air. Whether or not this assumption is exactly correct makes little difference inasmuch as the higher terms containing r^{-8} and r^{-10} contribute relatively little to the final result.

Regarding r_0 for water vapor, this would have to be allowed to vary markedly with temperature in order to obtain perfect agreement with observed data even in the range −10 to 80 C. The reason lies, of course, in the use of the dotted curve of Fig. 1 which is not an adequate approximation in the case of water with its strong permanent dipole moment. Ultimate refinement in the calculations with a view toward extrapolating experimental findings over the widest possible temperature range will be reserved for a later paper. Meantime a

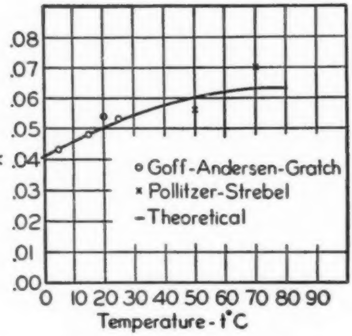

FIG. 3. COMPARISON OF CALCULATED AND EXPERIMENTAL VALUE OF λ

constant value $r_0 = 3.064 \times 10^{-10}$ cm adjusted to perfect agreement at 350 K will be used.

Following Fowler's[13] suggestion of averaging the r_0's *arithmetically* and using the combination rules explained above for determining the coefficients in Equation (6) for water-dry air mixtures (Row 7, Table 2), the interaction constant A_{aw} can be calculated as a function of temperature by means of Equation (5). Then from known values of A_{ww} and A_{aa}, the corresponding values of λ are computed. Fig. 3 shows the remarkable agreement thus obtained with the experimental values.

The calculated values of the interaction constant for moist air have been reduced to the following empirical equation,

$$A_{aw} = \frac{16{,}274}{T} \times 10^{43/T} - 4.25 \text{ (cc/mol)}$$

valid in the range −10 to 80 C, it being understood that T is in degrees Kelvin. Comparison with experimental values is repeated in Table 5.

[13] Statistical Mechanics, by R. H. Fowler. (Cambridge University Press, 1936, p. 308.)

TABLE 5—INTERACTION CONSTANT λ FOR MOIST AIR

T (C)	OBSERVED	CALCULATED
5	0.043	0.043
15	0.048	0.048
20	0.054	0.050
25	0.053	0.052

SUMMARY

Final values of the interaction constant for moist air for the range 5 to 25 C are reported. The experimental measurements from which these are derived have been critically evaluated to ascertain the limits of uncertainty in the final results. These limits of uncertainty were found to be sufficiently narrow to insure that subsequent calculations of the various thermodynamic properties of moist air based on the final values of λ will not be in error more than about ±0.03 per cent.

Considerable confidence in the permanence of the experimental values is afforded by the remarkable agreement with values calculated from recent quantum statistical theory. Further refinements in the calculations promise to provide a reliable basis for extrapolation to cover at least the range −100 to 200 F. It can be concluded from the theory and inferred from the fact that the high-pressure measurements of Pollitzer & Strebel[1] conform to the theory that the various thermodynamic properties of moist air can be calculated from present knowledge of the interaction constant for pressures possibly as high as 100 atmospheres.

ACKNOWLEDGMENT

The authors wish to express appreciation of the generous support given by the American Society of Heating and Ventilating Engineers in the prosecution of this cooperative investigation. They wish also to say that during the latter half of the investigation they found repeated occasions to admire the expertness of Dr. A. C. Bates, Assistant Professor of Mechanical Engineering, Lehigh University, in designing and building the apparatus with which the measurements here reported were made.

DISCUSSION

W. H. CARRIER, Syracuse, N. Y.: I am glad to hear Dean Goff say that Dalton's law is a rule. The fact that the pressures correspond to the volumes makes it a very convenient rule to use. I do not believe, however, that it has any standing. That is your real answer, is it not?

DEAN GOFF: Yes. The concept of partial pressures implies the existence of individual pressures which add up to the total pressure and which, together with the uniform temperature of the mixture, determine individual energies, enthalpies, and entropies, which add up to the total energy, total enthalpy, and total entropy respectively, of the mixture. With this understanding, partial pressures simply do not exist and Dalton's law becomes merely a rule to be used when no better information is available.

DR. CARRIER: In any mixture of two gases there is a definite ratio of the number of molecules of one gas to that of another in a given space, and it is assumed that the two gases mix uniformly throughout the space by diffusion. The socalled partial pressures are merely hypothetical or derived values conveniently used in calculations but in themselves do not have any primary physical significance. The same is true of a mixture of a socalled permanent gas with a saturated vapor. We say that for a given saturation temperature of a vapor there is a definite vapor pressure within the enclosed space. If the temperature is maintained the same and the pressure is reduced, the vapor will be superheated. If we attempt to increase the pressure by adding more vapor into the space and, at the same time, maintain the temperature constant, the vapor will condense and the pressure will not change. What this really means, according to my understanding, is that there are a certain maximum number of molecules per unit of volume that can exist in the vapor phase, which is another way of saying that for every temperature there is a possible minimum average distance between the molecules. According to the socalled rule of partial pressures, this average distance between molecules is the same whether another gas is placed in the mixture or not. This rule only applies when there is perfect diffusion of the two gases, one with the other, and this is in itself a relatively slow process. This can best be illustrated by the following experiment:

Take a tall vessel or long test tube and place in the bottom a liquid of high molecular weight, such as F-11 for example, (CCl_2F) which has a boiling point of 75 deg at normal atmospheric pressure. Insert it in a water jacket at 75 deg and exhaust all the air, at the same time boiling off a part of the liquid to fill the space above with vapor. Now introduce air into this vessel from the top slowly, perhaps through a fine mesh screen, so that it enters the top of the vessel without turbulence. If the temperature is maintained, the pressure will not increase in spite of the fact that the vessel is perhaps half filled with air at the top. Bear in mind that the diffusion between the air and this vapor is extremely slow, due to their difference in molecular weight. However, if the vessel now be sealed, the pressure will gradually rise as diffusion slowly progresses, and after many hours of standing the maximum pressure will be reached of practically the theoretical or 50 per cent greater than the initial pressure (assuming that 50 per cent by volume of air at the same pressure is introduced).

Not until this diffusion is completed, however, will Dalton's rule of partial pressures hold. Dalton's rule is contingent upon perfect diffusion and a partial pressure in no way accelerates diffusion, as is frequently assumed. It is only because pressure bears a relation to volume that we conveniently use the partial pressure ratios instead of the volume ratios.

Under certain conditions, all gases are considerably affected by what may be called interaction constants and, for this reason, show a considerable deviation from the theoretical gas law or equation of state. Water vapor at very low pressures follows very closely the theoretical gas law ($pv = RT$). Also in mixtures of air and water vapor at relatively low or atmospheric pressures, the interaction constant between air and water vapor is small.

DEAN GOFF: That is correct. Referring to the first comment as to the difference between our results now and what you would get by this most natural application of Dalton's rule, the difference is only about a half a per cent in the low pressure range, so that if you are not interested in half a per cent you could probably afford to forget it.

DR. CARRIER: By volume, or weight?

DEAN GOFF: A half per cent by weight at saturation. That is just a rough figure. We computed it for 68 F and atmospheric pressure. If to use best data involved any complication, then it would certainly be a practical thing to forget that half per cent. At a pressure of 100 lb per square inch, the discrepancy goes up to 15 per cent, and at 200 lb per square inch the discrepancy is of the order of 30 per cent, so it is fortunate we are concerned here with atmospheric pressures.

In addition my contention is that the accurate data are more simple to use than Dalton's rule, anyway. So there are substantial reasons then why we should have accurate data if we present them in the right manner.

JOHN JAMES, Cleveland, Ohio: I believe it would be very helpful if Dean Goff would illustrate how the results of this research investigation will alter the principal temperature lines. If this were done, I feel that members attending this meeting would have a little clearer concept of the value of this work and its practical significance.

DEAN GOFF: On a psychrometric chart, first of all, you have to decide what coordinates you are going to use, and in our opinion there is only one choice that you can make. That is open to argument, of course; but we base our reasoning upon the fact that in all applications of thermodynamics or engineering we have two fundamental propositions that none of us will wilfully or knowingly violate, and these are the law of the conservation of energy and the law of the conservation of mass.

In air-conditioning, when applying these properties, it is necessary to analyze the case of steady flow where the fluid moves through a duct. The air stream is continuously moving across some fixed section of interest. Then it immediately becomes important to know how much energy crosses that section per pound of fluid and that quantity has been given the fancy name of *enthalpy*. In the first instance it is important to learn with confidence that whenever a fluid passes a fixed section of duct or any place else, moving across that section, the energy it carries with it is its enthalpy, which is one of the quantities appearing in Table 6.

The next thing is the quantity *humidity ratio*. In any application it is necessary to account for the total weight of water crossing any section as well as the total weight of dry air, in order not to violate the law of the conservation of mass. The Society has used for years this quantity, the ratio by weight of water to dry air—the number of pounds of water per pound of dry air. Right now we are calling it the humidity ratio. The enthalpy and the humidity ratio are fundamental coordinates for the reasons cited and if left just as they are the psychrometric chart would take the form of a scroll, and would have to be unwound in order to use it.

Presently we have adopted the trick of reducing the enthalpy by a thousand times the humidity ratio, and using the *reduced enthalpy* as abcissa instead of the enthalpy itself. This simply amounts to a plotting on oblique coordinates and pulls the chart up into fairly decent proportions. It gets us into complications, and maybe we can think of a way out of that in the actual drawing of the chart, but this is what we are doing at the present.

One advantage of the chart is that these isotherms extend out into the region which we call the two phase region, where both liquid and vapor phases exist. The liquid consists mostly of pure water but has a little air dissolved in it. The water phase consists mostly of air with vapor mixed with it.

In the tower region there will exist three phases: ice, liquid, and vapor, and when you have this chart built up using the fundamental coordinates mentioned then the analysis of an air-conditioning problem usually involves nothing but straight lines which are very simple to follow on the chart. This we call the Mollier chart, originated in 1923 by Richard Mollier, who also originated the Mollier steam diagram.

C. M. HUMPHREYS, Pittsburgh, Pa.: I should like to add this comment: I do believe Dean Goff has taken what sounds like a very difficult subject and put it into very usable form. There is naturally going to be some resistance to changing over from our customary psychrometric chart to a Mollier diagram, but I am certain that if we take the time to investigate that diagram, we shall find that any air conditioning problem that can be solved on the psychrometric chart can be solved on the Mollier diagram, and usually can be solved more easily. I think he has done a wonderful job for the Society.

DEAN GOFF: I would qualify one statement by Professor Humphreys, and by qualifying it, make it perhaps a little stronger; that is, I will guarantee that any air-conditioning problem that can be solved by thermodynamics alone, where it is a thermodynamic problem, can be solved more simply on the Mollier chart than on any other chart. But, I would qualify it to say that it must be amenable to thermodynamic analysis. I am not sure of problems in coil selections where such mechanisms as heat transfer and material transfer are involved. I would not go so far as to make the guaranty there.

No. 1239

STUDY OF ACTUAL VS. PREDICTED COOLING LOAD ON AN AIR CONDITIONING SYSTEM

By James N. Livermore,* Detroit, Mich.

IN THE design of an air conditioning system the amount of refrigeration to be installed and the sizes of the fans, ducts and cooling coils are based on certain calculations of the expected cooling and dehumidifying loads. The methods of making these calculations have been developed partly from theory and partly from laboratory tests without the benefit of much actual operating data. The author has long believed that this is an undesirable situation—that there should be more frequent checks made between operating results and design figures.

This paper reports some tests of the air conditioning system in an office building in which the refrigeration load was measured and separated by measurement or calculation, into its various components. The test results are compared with the design calculations and the discrepancies are analyzed and discussed.

The building on which this study is based is illustrated in Fig. 1. It was designed for the particular uses of certain departments of The Detroit Edison Company and is arranged to house offices, drafting rooms, electric meter shops, etc. Six floors are above grade and one below, the plan of each being a full rectangle approximately 120 ft by 250 ft. All corridors and utility spaces are located in interior bays. In some respects this is a loft type of building since it has many large open floor areas with a minimum of partitions. It also departs from the conventional office building in that its exterior walls are built of solid panels of glass block and masonry, leaving the structure virtually windowless.

Generally the interior artificial lighting is a coffer type in which lamps are set in open recesses in the ceilings as shown in Fig. 2. These coffers are arranged in a uniform pattern of nine units per bay and each is fitted with a lamp of capacity adequate to produce 50 foot-candle lighting intensity at desk level.

The entire building is both heated and cooled by three air conditioning systems[1] centered in the basement. The refrigerating plant illustrated in Fig. 3 consists of two motor driven centrifugal machines designed for chilled water service. One machine has 360 tons of refrigerating capacity, and the other, 180 tons. The output of each is controlled by manual changes in compressor motor speeds.

Hot water for building heating is provided by a heat exchanger to which steam is supplied from the district heating mains.

* Mechanical Engineer, The Detroit Edison Co. Member of A.S.H.V.E.

[1] Unusual Air Conditioning Problems Met in Designing New Glass Block Building, by James N. Livermore. (*Heating, Piping and Air Conditioning*, April, 1938.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, June, 1943.

FIG. 1. THE DETROIT EDISON COMPANY BUILDING HOUSING OFFICES, DRAFTING ROOMS, ELECTRIC METER SHOPS, ETC.

The air handling system is a double duct type and is arranged as shown diagrammatically in Fig. 4. Warm and cold air supply shafts extend in pairs the full height of the building in three central locations. Mixing dampers at each floor proportion warm and cold air from these shafts to supply the cooling or heating demands of thirteen or more control zones on each floor. At the base of each shaft is a supply fan. As the demand for cold or warm

FIG. 2. INTERIOR VIEW SHOWING LAMPS SET IN OPEN RECESSES IN CEILING

air varies, the static pressure is held constant on each shaft by controls which vary the position of fan inlet dampers. This is supplemented by manual changes of fan speed when required. From the central refrigerating plant chilled water is supplied to the cooling and dehumidifying coils at the three cold shaft fans and likewise from the steam heat exchanger hot water is supplied when required to the heating coils at the three warm shaft supply fans. Although the total quantity of air being supplied remains approximately constant at all times, it may be seen that the quantity of air through any one supply shaft, either cold or warm, will be continually varying with the demands of the mixing dampers it supplies.

Supply air leaving the mixing dampers is carried through individual zone ducts to diffusing areas in the room ceilings. A greater part of the building

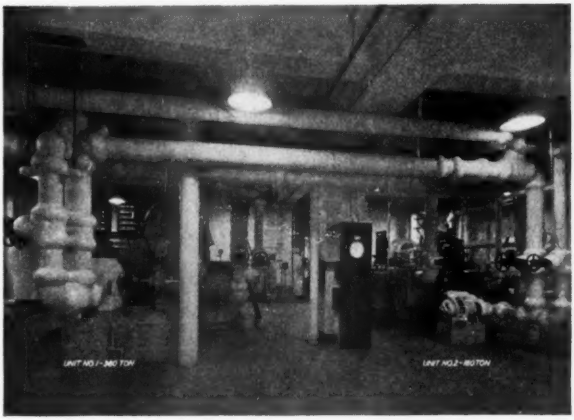

FIG. 3. MOTOR DRIVEN CENTRIFUGAL MACHINES DESIGNED FOR CHILLED WATER SERVICE

has perforated metal acoustic ceilings and portions of these are utilized as a means of introducing supply air to the occupied spaces. Air leaving the rooms travels to the main return shafts either through openings in the window sills and then through the ceiling space of the floor below or through corridors direct to the return shafts.

In addition to the mixing damper, each control zone is equipped with a remote bulb thermostat arranged with the sensitive element in the office space, but with the instrument proper out of reach of the occupants. Each of these thermostats, 82 in all, is connected to a readjusting control known as the *reset* which is capable of automatically changing the setting of each in response to changes of outdoor temperature.

PROCEDURE

In order to obtain significant data on the performance of this air conditioning system during the cooling season, it was important to conduct tests during

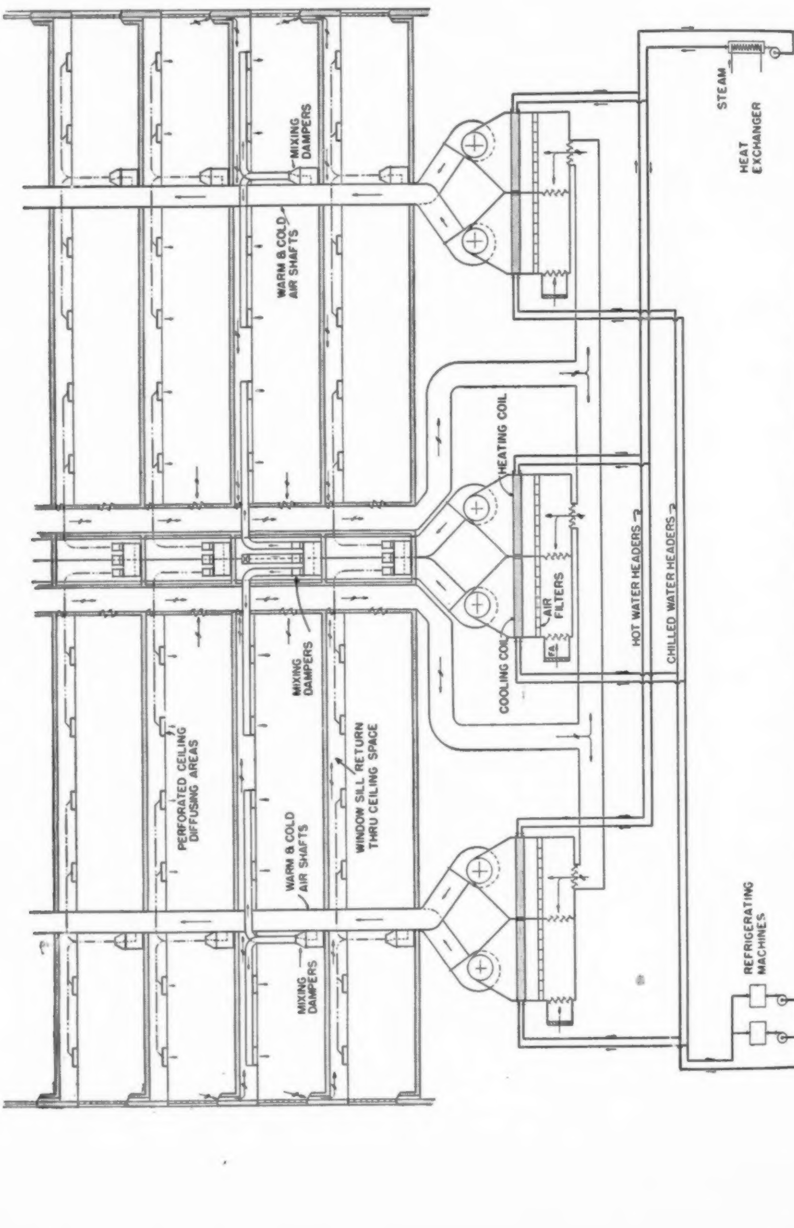

FIG. 4. DIAGRAM OF DOUBLE DUCT TYPE AIR HANDLING SYSTEM

weather approximating the conditions assumed as a basis for its design. In an effort to do this much unsuitable data was gathered, since many test runs were started only to have the following weather turn too cool for heavy load operation. Generally, the testing was done by three four-man crews working in shifts independent of the operating force, and the tests were run continuously for 48- and 72-hr periods.

Specifically, the object of this study was to measure the total cooling load as accurately as possible and to simultaneously obtain data on its components. In addition to the total load, therefore, data were to be collected on the load

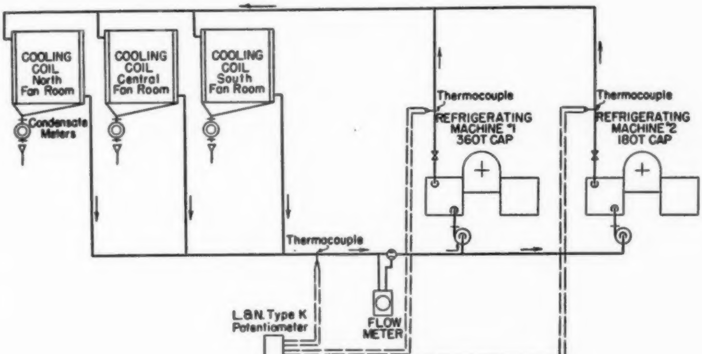

Fig. 5. Arrangement of Test Apparatus on Chilled Water Circuit

due to the following: (1) Solar radiation and heat transmission through roof and walls; (2) Electric light and power; (3) Occupants; (4) Ventilating air.

Fig. 5 shows schematically the arrangement of the apparatus used in measuring the total refrigerating load. This provided data for calculating the rate of heat *removal* from the chilled water as it passed through the refrigerating machines. Due to the equipment layout this was a simpler method than taking data on the rate of heat *gain* at each of the three main fan room cooling coils.

Generally the apparatus consisted of a means of measuring the chilled water flow and the temperature drop which took place. Since it was never necessary to operate both machines simultaneously, one flow meter and one temperature reading on the return side of the plant was sufficient. A recording flow meter had originally been installed in the chilled water piping which was to meter the total flow through the two machines. The accuracy of this instrument was established by water leg test on the recorder, and conformity of the nozzle design and installation to well established standards. The meter was also checked against the main building city water meter and the two were found to be within 1.5 per cent in agreement. Having accurate flow data, it was necessary only to measure the temperature drop of the water through the refrigerating plant to arrive at the total rate of heat removal.

The accuracy of water temperature readings was very important since each

degree Fahrenheit drop represented approximately 60 tons of refrigeration when the large machine was running and 30 tons when the small machine was running. It was felt that readings with error of no greater than 0.1 F were necessary. To obtain this degree of accuracy, thermocouples were used in conjunction with a precision type potentiometer. To avoid temperature lag the thermocouples were installed in contact with the water. For their correct location in the pipe, a preliminary traverse was made, taking several readings over one cross section. These indicated that a single reading at the center of the pipe would give a fair average temperature indication and that the effects of stratification in the pipe could be neglected. Since the readings of water temperature in and out of the refrigerating machines could not be made simultaneously, special care had to be taken to observe a true difference

Table 1—Typical Data Showing Temperature Readings for One Load Computation

Time	Supply Temp. F	Chilled Water Temperature		Ave. Temp. Drop F	Chilled Water Flow
		Return Temp. F	Temp. Drop F		Gpm
9:00 A.M.	43.8	49.2	5.4		810
	44.1	49.4	5.3		810
	44.3	49.7	5.4		810
	44.3	49.9	5.6		810
to	44.5	49.9	5.4	5.4 F	810
	44.6	50.1	5.5		810
	44.7	50.2	5.5		810
	44.8	50.2	5.4		810
9:10 A.M.	44.8	50.2	5.4		810

between them. Thus, each observation consisted of a number of readings taken alternately between supply and return for a period of 10 min at approximately 30 sec intervals. Typical data taken for one load computation are shown in Table 1.

Solar Heat Gain

In early test runs corresponding indoor-outdoor surface temperatures were measured with a view to approximating a rate of heat gain through the building walls. Proper interpretation of the data, however, became so complex that it was decided to repeat a portion of the tests during the next cooling season getting complete data on solar intensities.

The following summer, therefore, an Eppley pyrheliometer was used to record continuously the intensity of the sun radiation perpendicular to a horizontal plane at roof level. Having this information, hourly intensities normal to the walls of the building were then computed, and used with the new A.S.H.V.E. heat gain data [2] collected in Pittsburgh during the same season. In order to make proper allowance for diffused radiant energy or

[2] A.S.H.V.E. Research Report No. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson, and Burt Gunst. (A.S.H.V.E. Transactions, Vol. 46, 1940.)

solar sky reflection to the shady sides of the building, empirical values appearing in the same paper are used in this report for purposes of load analysis. These values were spot checked but no continuous readings were made during tests. In order to make a proper allowance for the effect of Venetian blinds, a survey was made on a bright sunny day, and a detailed hourly record kept of the areas of glass block shaded by the Venetian blinds in each day. The data thus obtained were then assumed to be representative shading for any day when sun was a load factor.

Electric Light and Power

The interior load due to electric lighting and miscellaneous electrical devices throughout the building could not be measured directly. However, it was possible to read the total hourly electrical input to the building on the main kilowatt-hour meter. Also for the purposes of this test an electric meter was installed on every motor of 3 hp or over. The difference between the total building power and the total motor power then gave the kilowatt-hours consumed for lighting. The correctness of this method was checked by having a test crew actually observe the number and size of every lamp in use in the building each hour for several hours while another crew read the electric meters. This direct observation of energy consumed by lights agreed very closely with that measured by meter difference.

The majority of the motors referred to are located either in elevator penthouses or in the basement machine room, both of which are separated from the rest of the building and are not air conditioned. These data made it possible to exclude such motors in the heat source analysis unless they did contribute to the air conditioning load. The power consumed by chilled water pump and supply fan motors was itemized separately as a source of cooling load after a deduction had been made for motor losses.

Occupant Load

Since the number of occupants in the building varied from hour to hour through the day, but not from one day to another, a single survey of these was judged sufficiently accurate for the purposes of this test. Therefore, an actual count of occupants was made each hour throughout the building on a weekday. These data are shown in Table 2. Saturdays and Sundays the building occupancy was so low that it was neglected as a source of load.

Ventilating Air

The sensible and latent heat load due to outdoor air drawn into the fan room intakes is a function of rate of air intake and the difference between the dry- and wet-bulb temperatures of outdoor air and recirculated air. Data for these

TABLE 2—HOURLY SURVEY OF NUMBER OF OCCUPANTS

TIME	2 TO 7 A.M.	8 TO 9	9 TO 10	10 TO 11	11 TO 12	12 TO 1 P.M.	1 TO 2	2 TO 3	3 TO 4	4 TO 5	5 TO 6	7 TO 10	11 TO 1 A.M.
Number of Occupants....	20	701	671	615	460	477	630	636	617	334	38	10	25

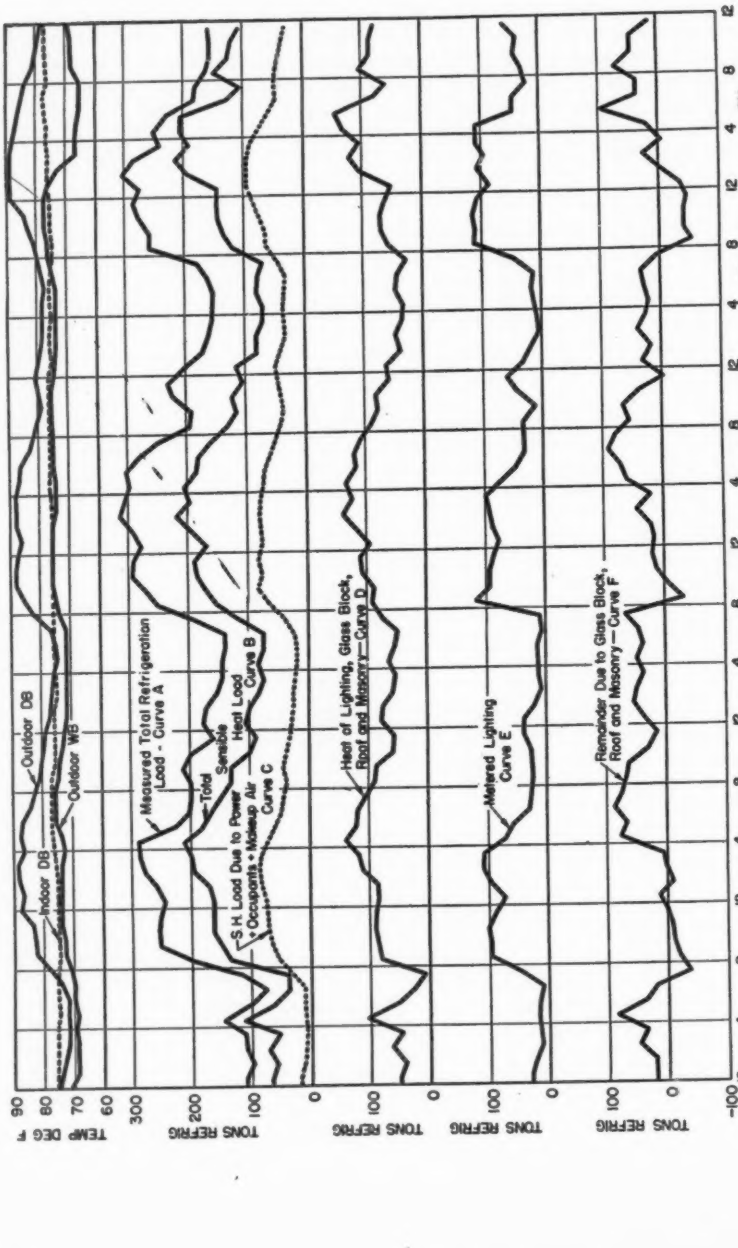

FIG. 6. COOLING LOAD ANALYSIS (AUGUST 2, 3 AND 4)

calculations were taken hourly. The air quantity had a tendency to vary and to facilitate measuring its rate of flow, baffles were installed in each intake to serve as metering orifices with manometers. Several rates of flow through these orifies were then measured with anemometers and the corresponding pressure drop through the orifices as indicated on the manometers was recorded. This gave data for a calibration curve for each manometer so that its readings could be used to determine the rate of flow readily.

Latent Heat Load

All moisture precipitated from the three cooling and dehumidifying coils was collected and carefully measured at the end of each hour of operation. With these data it was possible to calculate accurately the over-all latent heat load on the plant, this method completely avoiding the errors of instrument reading often involved when wet-bulb temperatures are used.

RESULTS

Analysis of Cooling Load

From the mass of data taken, the most typical of heavy load operation seems to have been secured during a 72-hr test made in early August. The uppermost curves of Fig. 6 show the temperature conditions during this run. It will be noted that at no time did the dry-bulb rise above 89 F, although on all three days the wet-bulb ran to 75 F and above, contributing greatly to the loads indicated. It is important to notice here that during this test the average building temperature was held fairly constant at 75 to 76 F from 8 a.m. to mid-afternoon, at which time it was allowed to rise. This practice has been adopted with a view to reducing temperature shock to employees on leaving the building at the end of the day.

Curve A is a plot of the measured total heat removed from the building by the refrigerating plant. The minor irregularities of the curve are largely due to manual changes in compressor speed and the consequent effect of allowing the temperature level of the chilled water to fluctuate slightly. The major trends of *Curve A,* however, are due to variation in load sources imposed upon the plant and an attempt to evaluate these is shown in the curves below.

Curve B is a plot of the total sensible heat load on the plant. These hourly values were accurately established by subtracting from the total cooling load, the latent heat of condensation of all the moisture collected from the cooling coil surfaces for each corresponding hour.

Curve C is a total of the heat due to fan and pumping power, plus the sensible heat given off by occupants, plus the sensible heat removed in bringing ventilating air from its outdoor dry-bulb temperature down to that of recirculater air. There are doubtless some inaccuracies in these values due to the method of counting occupants and a certain margin of error is to be allowed in measuring the air flow through the outdoor air intakes. However, the magnitude of these errors is believed to be less than 10 per cent of the totals represented by this curve.

If, from the total hourly sensible heat loads shown in *Curve B,* corresponding values in *Curve C* be subtracted, the remainder will leave heat values which can be attributed only to building illumination together with radiant and con-

ducted heat passing through the building enclosure. An hourly plot of this remainder produces *Curve D*.

Curve E is an independent plot of the heat equivalent of the hourly electrical energy consumed in lighting the building. If these values now be subtracted from corresponding values in *Curve D* the remainder, shown in *Curve F*, can be attributed only to the heat gain through the glass block walls, the roof, and

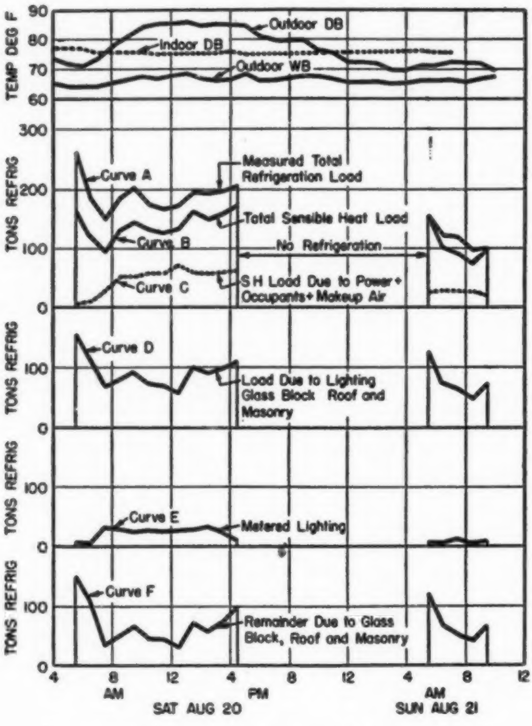

FIG. 7. COOLING LOAD ANALYSIS (AUGUST 20-21)

masonry around columns and floor spandrels, plus whatever errors may be involved in test measurements.

A brief inspection of corresponding indoor-outdoor temperature conditions will show that many of the heat flow quantities indicated in *Curve F* are quite obviously in error. On August 2 and 4 the values indicate (of course erroneously) that heat flowed *outward* through the walls from 7:00 a. m. until noon or after. While no data were taken on these days to permit close calculation of the rate of heat gain through the glass block, this material predominates in the wall areas, has a very short heat flow time lag, and the existing temperature differential and solar radiation would have made the indicated outward

heat flow impossible. Other tests showed similar results when subjected to the same analysis with one notable exception: Fig. 7 shows a similar breakdown of the sensible heat load on a Saturday and Sunday when electrical energy used for lighting was at a minimum. On these two days no negative values were indicated for heat gain through the building enclosure even though the outdoor temperature was lower than on August 2, 3, and 4. Since the time when lighting was a minor heat source this sensible heat analysis gave more reasonable values for wall and roof load, there was a definite indication that the effect of lighting had been overestimated. While the heat equivalent of the kilowatt-hour cannot be challenged, the time required for the increase of one kilowatt-hour of heat released in a building to be felt by the cooling plant remains to be determined.

Sturrock[3] finds that approximately one-third of the total energy input to an incandescent lamp is the maximum that can be carried away directly by circulating air over it, and that the balance which is entirely radiant energy must be first intercepted by some heat absorbing surface before it can affect the air temperature. Since in this case the lamps are set in coffers and are not subject to any artificial circulation, it is probable that an even smaller portion of this heat *immediately* affects the room air temperature. This is not to argue, of course, that the total energy input to each lamp does not ultimately become a part of the building cooling load (neglecting the small amount which may be radiated to the outdoors). It does seem probable, however, that the lamp heat effective at the refrigerating plant has a far different hourly load curve than that represented by the metered energy in *Curve E.*

In *Curve F,* Fig. 6, the greatest *negative* values for wall and roof load appear simultaneously with the beginning of heavy lighting load and diminish throughout the day until heavy lighting ceases. This gradual change seems to definitely indicate that some lighting heat is being stored up and being released hours later. In view of these considerations it seems doubtful if the heat due to lighting can be treated simultaneously and at *full value* with other known heat sources, thus arriving at correct values for *Curve F.*

This failure to reconcile the total measured load with the sum of its parts seemed very important, particularly when all heat loads had been very carefully measured with the exception of that through roof and walls. Since this discrepancy might lead to certain changes in the calculation of cooling loads generally, a repetition of the test, obtaining at the same time data for accurately estimating the heat gain through the building enclosure, seemed well justified.

The analysis of a continuous 48-hr test, made through July 7 and 8 of the following year, is shown in Fig. 8. Although the outdoor dry- and wet-bulb temperatures were somewhat lower on the second day than on the first, the curves of solar intensities for the two days, which were virtually cloudless, are practically identical. Since July 8 was a Saturday, the building was practically vacant and a large difference between the two days is shown for load due to lighting and occupants.

Operating practice during this test had changed from that of the year before, particularly in the control of indoor temperatures. Where previously the setting of all room thermostats was varied automatically in response to changes of outdoor temperature, this method was abandoned in favor of resetting the

[3] Effects of Artificial Lighting on Air Conditioning, by Walter Sturrock. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938.)

thermostats manually from a remote central control point. Indoor temperature adjustments made by the operator had a striking effect on the load at the refrigerating plant as reflected in *Curve A*, Fig. 8. Here we see a continuous decrease in total load from 9:30 to 11:30 a.m. on July 7, even though the

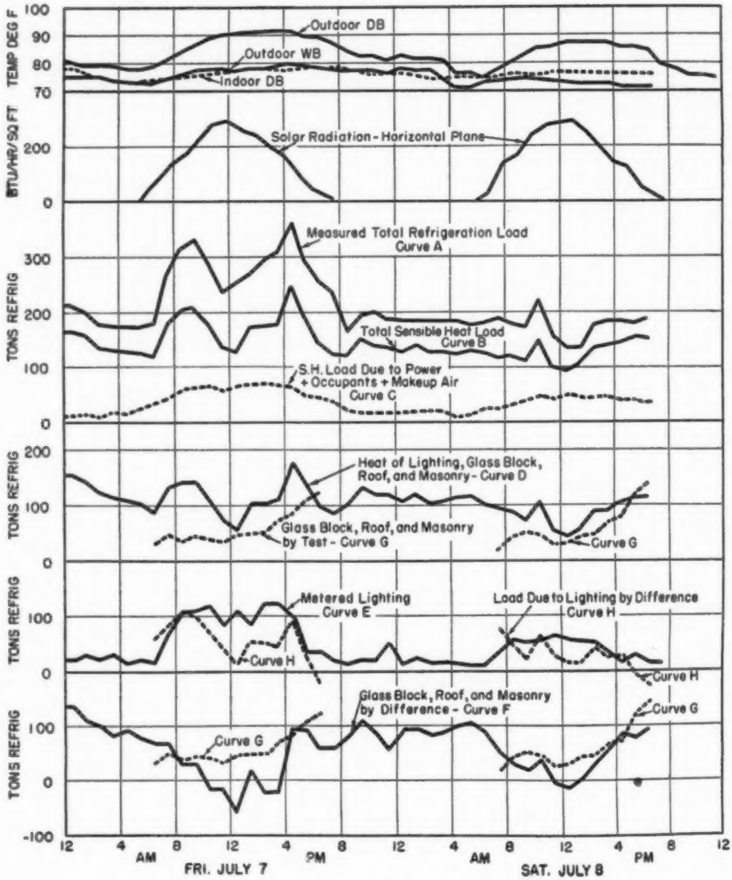

FIG. 8. COOLING LOAD ANALYSIS (JULY 7-8)

outdoor temperature rose from 86 to 90 F during the same interval. At the start of this period the building temperature was 75 F. From time to time during the morning hours the operator raised the thermostat settings, thus allowing the average temperature to rise approximately one degree. This permitted much of the interior heat to be absorbed by interior materials which

were at a temperature lower than the new thermostat settings, and, in effect, unloaded the refrigerating plant. During the rest of this day (until 7:00 p. m.) the thermostat settings were left unchanged, and *Curve A* shows loads corresponding to constant temperature operation.

The procedure for analysis of the cooling load is much the same as that explained for Fig. 6. *Curve A* is the total cooling load for the building. *Curve B* is the total sensible heat load. *Curve D* is the remainder after the sensible heat load due to power, occupants and make-up air have been subtracted from *Curve B*. *Curve G* is heat gain through glass block, masonry and roof calculated on the basis of solar intensities observed during the test and with benefit of the latest information on the subject as it appears in the Heating, Ventilating, Air Conditioning Guide, 1942, pp. 141-149.

If the values in *Curve G* be subtracted from corresponding values in *Curve D* the remainder must represent the load due to the heat of lighting. Plots of this remainder appear in *Curve H*. These are to be compared to *Curve E* which represents the actual metered lighting load.

There are two explanations for the differences in the cooling load due to electric lighting indicated by these curves. The first is the likelihood of temporary storage of heat from incandescent lamps in the building material as previously suggested. Secondly, with the manipulation of the thermostats in the hands of the operator, it is possible for him to vary the load on the cooling plant without regard to weather conditions, *i.e.*, by raising the control settings, the plant is temporarily unloaded, and by lowering the settings the plant may be loaded beyond its capacity. This manipulation of controls plus cumulative errors in the data could easily account for the erratic relationship between the two curves. The significant part of the data then is that with all other heat sources carefully measured and computed, the heat due to lighting during the course of a day averages considerably less than the heat due to electrical input. In this case, on both July 7 and 8 the remainder due to lighting (*Curve H*) is 36 per cent less than the electrical input.

Curve F is of interest only to show again that if lighting load is taken at full instantaneous heat value, leaving wall and roof load as a remainder, the latter results in negative values just as in preceding tests, and the greater the lighting load the more extensive are these false remainders.

Comparison Between Design and Test Conditions

None of the test work was done when weather conditions exactly duplicated those assumed as a basis of design. However, a comparison between the operating conditions that were assumed at the time of design, and those which existed at the time of the test have a definite bearing on the findings of this investigation. It should be noted here that while the original calculations indicated that no less than 450 tons of refrigerating capacity would be required, at no time in the five years of operation has the cooling load been more than the 360-ton machine could carry.

In arriving at the capacity of the refrigerating plant, many assumptions were made as to the ultimate use of the building, and calculations were based on data which have since been considerably changed.

The following briefly compares the load components as they were originally calculated, with the same components using present data and corrected assumptions:

Glass Block: The original data on the solar heat transmission through glass block were given by the manufacturer as 54 per cent of an equal area of single glazed factory sash and calculations were made on that basis. New data on this material in the A.S.H.V.E. Guide 1942 would result in the following differences between original and present calculations:

	Heat Gain Tons				
	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original	78.0	52.4	79.0	105.5	79.0
Present	65.6	52.6	56.9	72.8	85.7
Surplus	12.4	−0.2	22.1	32.7	−6.7

Masonry and Roof: New data have been published on the time lag involved in transmitting heat through roof slabs and masonry. The original calculations were based on data[4] obtained by Sanford, Walker and Wells which showed a fairly constant rate of heat inflow for heavy masonry walls. Roof slabs were figured for a constant temperature gradient. The following difference would result from original and present data:

	Heat Gain Tons				
	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original	19.2	19.2	19.2	20.8	20.8
Present	8.6	8.5	21.7	29.9	33.9
Surplus	10.6	10.7	−2.5	−9.1	−13.1

Lighting: Original calculations assumed that 2700 watts of lighting would be installed in each bay, while in many locations this has been reduced to 1800 watts.

	Heat Gain Tons				
	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original	119.6	119.6	119.6	119.6	119.6
Actual use	112.0	86.3	86.5	125.0	36.2
Surplus	7.6	33.3	33.1	−5.4	83.4

It was also assumed that 40 per cent of the lighting would be continuously in use throughout a bright day. Comparison between this assumption and typical metered use is shown in last tabulation in left-hand column.

If, as the data in this test indicate, the *effective* heat equivalent of lighting averages to be 36 per cent less than the heat equivalent of electrical input, the comparison becomes:

	Heat Gain Tons				
	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original	119.6	119.6	119.6	119.6	119.6
Actual	71.6	55.2	55.3	80.0	23.2
Surplus	48.0	64.4	64.3	39.6	96.4

[4] Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford, and E. P. Wells. (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1932.)

Occupants: An ultimate building population of 1800 was allowed for. Although the entire building was in use at the time of the test, actual hourly count showed a maximum of 701 persons indoors at any one time and this number fluctuated widely. The comparison is as follows:

	Heat Gain Tons				
	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original	60.8	60.8	60.8	60.8	60.8
Present	22.5	15.4	21.1	20.7	1.3
Surplus	38.3	45.4	39.7	40.1	59.5

The loads due to ventilating air and power for fans and pumping do not differ materially from those originally predicted.

These comparisons are summarized in Table 3, the loads in both cases being based on the following schedule of indoor-out-door temperatures both for the original and present calculations:

Time	Outdoor		Indoor Dry-Bulb
	Dry-Bulb	Wet-Bulb	
9:00 A.M.	80.0	71.5	75.0
11:00 A.M.	86.1	73.5	78.2
1:00 P.M.	93.6	74.8	79.4
3:00 P.M.	95.0	75.0	79.4
5:00 P.M.	95.0	75.0	79.7

Indoor air dewpoint constant at 57 F.

EFFECT OF FLUCTUATIONS IN CHILLED WATER TEMPERATURES

At the time the system was designed, it was not known how close the control of temperature of the chilled water supplied to the cooling coils would need to be to assure satisfactory performance. Therefore, chilled water temperature controls had been specified as required refrigerating machine accessories. These devices depended for their operation on varying condenser water flow in response to slight changes in chilled water temperature. Such an arrangement is not economical of condenser water, which is important in this case since city water is used, but rather allows a wide variation in condenser water discharge temperature for any one compressor speed. An alternative, though not close, control of chilled water temperature can be accomplished with these machines by varying the compressor speed and manually adjusting the condenser water flow to some constant rate which will keep its discharge temperatures within a desired range. Better over-all operating economy could be obtained by this method and Fig. 9 shows to what extent its adoption affected the conditions in the building generally.

The rate of latent heat removal by the cooling coils varies with changes in any of several factors. Two of these factors, the dewpoint of the air entering the coils and the rate of chilled water supplied during this test were held fairly constant at the values noted. The changes in two other factors affecting coil performance are shown in the curves, namely, temperature of chilled water

TABLE 3—COMPARISON OF LOAD CALCULATIONS—ORIGINAL VS. PRESENT

TIME	9	11	1	3	5
ORIGINAL CALCULATIONS	LOAD IN TONS				
Glass block	78.0	52.4	79.0	105.5	79.0
Masonry and roof	19.2	19.2	19.2	20.8	20.8
Lighting	119.6	119.6	119.6	119.6	119.6
Occupants S.H.	34.0	34.0	34.0	34.0	34.0
L.H.	26.8	26.8	26.8	26.8	26.8
Ventilation S.H.	17.3	27.4	49.3	51.0	51.7
L.H.	72.2	79.5	67.2	66.2	60.5
Power	26.5	26.5	26.5	26.5	26.5
Total Load	393.6	385.4	421.6	450.4	418.9
PRESENT CALCULATIONS					
Glass block	65.6	52.6	56.9	72.8	85.7
Masonry and roof	8.6	8.5	21.7	29.9	33.9
Lighting	112.0	86.3	86.5	125.0	36.2
Occupants S.H.	12.6	8.6	11.8	11.6	0.7
L.H.	9.9	6.8	9.3	9.1	0.6
Ventilation S.H.	17.3	27.4	49.3	51.0	51.7
L.H.	72.2	79.5	67.2	66.2	60.5
Power	20.8	17.6	23.3	22.4	24.0
Total Load	319.0	287.3	326.0	388.0	293.3

entering and leaving the coils and the rate of flow of air through them. With this system, the demand for cold air increases in response to a rise of room temperature above the thermostat settings; therefore the sensible heat removal curve indicates the variation in air flow through the coils. The intervals between changes of compressor speed are also shown.

The most important conclusion to be drawn from this comparison of data is that the chilled water temperatures do *not* have to be closely controlled in order to produce satisfactory room conditions. In this case, the water temperatures varied for short periods between levels as high as 54 F and as low as 40 F with little effect on the dewpoint of return air, *i.e.*, the average moisture content of the air within the building.

EFFECT OF SHUTDOWN

Generally it has been necessary on week days to maintain comfort conditions throughout the building until 1:00 a. m. to accommodate the night force, rather than shutting down at 5:00 p. m. when the offices are closed. These long hours of operation were not expected originally. For purposes of study, however, data were taken on loads encountered after an all-night shutdown lasting from 5:00 p. m. until 5:30 a. m. These data are shown graphically in Fig. 10. None of the shutdown periods shown extend through extremely warm weather and so do not show the longest time that might be required to attain comfort conditions when starting up again. Further, the curves of indoor dry-bulb temperature are for the recirculating air and, thus, represent the building average. While these show a maximum rise of 1½ F during a shutdown period, some zones on the top floor and on south exposures rose to temperatures as high as

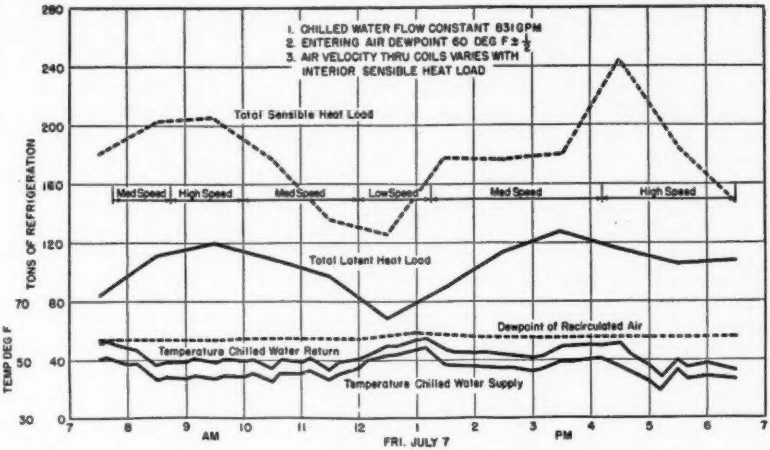

Fig. 9. Effect of Variable Chilled Water Temperature on Dewpoint in Occupied Space

85 F. More striking are the changes in wet-bulb temperatures and corresponding relative humidities of the return air which took place during shutdown. In one period the relative humidity rose to 63 per cent. This increase of moisture is not due to infiltration but rather due to continuous operation of the toilet exhaust system, which ultimately gets its air through the fresh air intakes. Since the building is practically air tight, this small amount of ventilation is desirable even though the building is not in use.

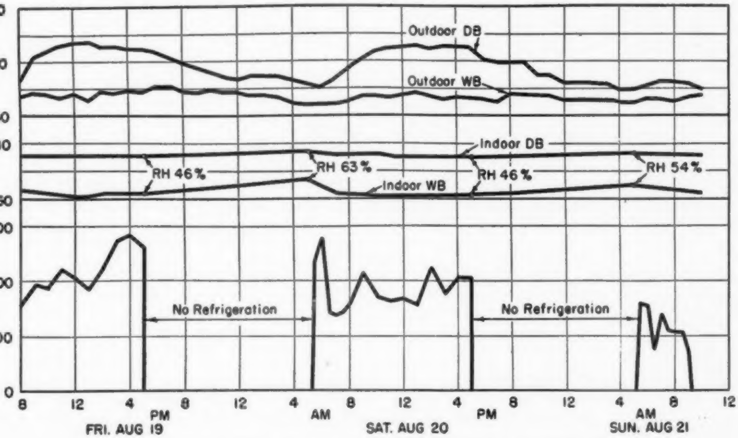

Fig. 10. Cooling Loads Resulting After Shutdown

On both days shown in Fig. 10 satisfactory indoor comfort conditions were attained within a period of two hours after starting the air conditioning equipment. Other operating records show that in extremely hot weather, after a week-end shutdown, as much as eight hours may be required. It has also been found necessary to start the refrigerating plant during shutdown periods for dehumidifying purposes any time the indoor relative humidity rises above 50 per cent. This is done for the benefit of drafting and mapping departments where wide moisture changes seriously affect drawing and blueprinting materials.

No exact relation can be established between the extra ton-hours of refrigeration required for cooling down after shutdown, and the ton-hours that would be used in maintaining comfort conditions through the same shutdown period. Figs. 6 and 8, however, give some idea as to the nature of the night load when operating continuously night and day.

Conclusions

The following may be concluded from this test:

1. The importance of well considered design assumptions cannot be over stressed. Faulty assumptions may completely offset all improvements to data. Some of these assumptions, those that pertain to the *use* of the building, such as the number of occupants, are the responsibility of the owner of the building. Others are the responsibility of the designer. It is important also that *safety margins* in assumptions shall not be allowed to become cumulative thereby causing a gross over-estimate of load.
2. Due to the assumptions and approximations that have to be made before calculations can begin extremely refined calculations for total cooling load are not justified.
3. In calculating the cooling load due to lighting, there should be a time lag allowance for heating up of building materials, with a varying proportion of the energy input going directly to the atmosphere. It is probable that where lights are surrounded by coffers as they were in this case, this time lag effect is accentuated. Further research on this point would be highly desirable.
4. Loads calculated for test conditions showed an erratic relation to those measured by test. However, the calculated peak loads were generally higher than the measured.
5. All loads seem to show a certain amount of lag or flywheel effect which is evidenced by the large amounts of heat to be removed at night when the plant is operated continuously.
6. Close control of temperature of chilled water to cooling coils is not necessary provided deviations from the required mean are not of long duration.

DISCUSSION

Thomas Chester, Detroit, Mich. (Written): In any type of engineering it is desirable to ascertain how results compare with planned performances, as this is full scale research. Consequently the analytical tests set forth in detail in this paper are interesting and useful.

It is reported that during the morning of July 7, the thermostat settings were progressively raised, which resulted in partial unloading of the refrigerating equipment, due to heat absorption by the contents of the building and the interior surfaces of the building itself and also the contained air, in rising to a higher temperature. The introduction of this variable seems a procedure of doubtful value in making performance tests.

However, it might be interesting to make a separate investigation along this line under stable load conditions, as on a cloudy day with approximately uniform outdoor temperature, preferably on a Sunday when there would be little fluctuation in the number of occupants, and with a constant lighting load. This would show the rate

of heat exchange between the fabric of a large office building and its air contents and also the rate of moisture exchange. The latter would be new but we have somewhat of a line on the former, indicated in the time required to heat a building in winter.

It is inadvisable to accept without proper understanding the statement that variations in water temperature for short periods between 54 and 40 F produced little effect on the average moisture content of the air within the building.

It should be realized that this pertains to this particular job and is not typical of air conditioning installations in general.

The building is large, it is unusually tight against infiltration, and the number of occupants relatively small. Consequently, the moisture load is small, as it is confined to the water vapor given off by the occupants and any residual excess after conditioning in the water vapor content of the outdoor air brought in for ventilation over the water vapor content of the indoor air. Therefore, the gain in the water vapor content of the indoor air would be slow.

Also, the air conditioning is done by coils. The performance would be quite different with an air washer type of conditioner, with which a rise in water temperature would immediately produce a higher dewpoint in the air leaving the apparatus. Even with coils, the gain in the moisture content of the indoor air would be at a more rapid rate in the case of a theater, in which the number of occupants per 1,000 cu ft of space would be much greater than in the case under notice, and hence, the relative and absolute humidity would rise more rapidly.

As stated, no general inference should be drawn from the reported slow rate of change in the moisture content of the indoor air with changes in supply water temperature, as the type of moisture load is the criterion of principal importance.

R. E. HATTIS, Chicago, Ill. (WRITTEN): I notice that the building walls have a considerable amount of glass brick. Was enough artificial light saved in the spaces near the glass brick walls to compensate for the increased cooling required? On future work would you consider a considerable reduction in the amount of glass brick wall, or even its entire elimination?

I would be interested in knowing whether water was used on the roof to reduce roof sun load, and whether the author would consider it on new work.

I would also like to know about operation of the cooling system with all recirculated air during all or part of the night to absorb part of the stored heat in building construction. If the cooling system was operated for only a part of the night, would it be operated immediately after working hours or just before the starting time the next morning?

The phenomenon of time lag in the heat from lights is interesting and, as the author suggests, is worthy of further research. This paper gave me considerable food for thought, and covered the subject in a comprehensive and interesting manner.

J. H. WALKER, Detroit, Mich. (WRITTEN): It is not often that the designer of an engineering project displays his calculations alongside of performance figures; nor in fact is it often in heating and air conditioning that any detailed performance tests are made. On both of these points the author deserves commendation.

The paper brings out at least two points which are impressive. The first one is that there may be an important time lag in the *internal* cooling load. We have always recognized, and made efforts to calculate, the time lag of heat flow through walls and roofs; but here is an apparent time lag in the effect of the internal lighting load. At any rate that explanation seems best to fit the test results. How unique that time lag effect is to this particular building design or whether it might be expected in other buildings cannot of course be stated. More work of a similar nature in other buildings is certainly needed.

Secondly, the paper shows how important are the design assumptions which must be made regarding such matters as the number of occupants, number of lights in use

at the time of the peak cooling load, etc. Reasonable accuracy in those respects may be far more important than refinement in thermal calculations.

More studies of this sort are highly desirable in order that design methods may be improved.

W. H. CARRIER, Syracuse, N. Y.: I consider papers of this kind highly important for the Society. This paper is a thoroughly comprehensive study of actual cooling loads compared with carefully estimated loads. This is very difficult data to get since it must cover a considerable period of time in order to include all the conditions on which the maximum load was based. It is further complicated by the temperature lag effects in the building structure which involves time as well as temperature conditions.

The engineer and the air conditioning manufacturer have occasion to make such tests only on rare occasions when the guarantees are not met or when special adjustments have to be made in order to meet the guarantees due to the actual margins being very small. Ordinarily the margins are perhaps more than ample so that considerable excess of cooling capacity, both in the air supply and in the refrigerating plant, is available and the plant "walks away" with the load. Under these conditions there is usually little incentive to be critical of the design of the installation so far as capacity is concerned.

Apparently in the installations studied by Mr. Livermore the refrigeration capacity was excessive. This was no doubt due to two factors—first, an over-estimation of the internal load due to occupants and lighting which was the client's fault rather than the engineers, and second, an over-estimation of the *instantaneous* value of the sun load. This was due to lack of adequate data on which to predict such loads and is one of the factors which needs study by the Society in order that engineers may make estimates that are less excessive. Apparently considerable of the heat that flows through the windows as sunlight is absorbed by floors, interior furnishings, etc., and is again re-radiated over a considerably increased period of time so that both the total load may be that estimated and the instantaneous load considerably less than that estimated. It is similar in character to the observed lag in heat transmission through walls on which the Society has been working for the past few years.

Probably the most satisfactory way to approach this subject is to provide sufficient cooling or heating to take care of the net heat gain or loss in the building during 24 hours with a suitable margin to be allowed for the maximum load as compared with the average load. In cooling a building it is generally not desirable to run for 24 hours. Hence, the net heat gain to the building in this period of time must be taken care of during 10 or 12 hours of operation. Very frequently the air cooling load, i.e., the net refrigeration effect of the air introduced into the building, is as great in the morning as any other time of day. This is particularly true of buildings of massive construction where there is considerable temperature lag in the walls. The building actually heats up over night and must be cooled down before occupancy. This may require anywhere from one to four hours of full operation. The refrigeration load may not be so great at this time for the reason that the outside air taken in for ventilation is not at its maximum wet bulb temperature. An especially heavy cooling load is usually encountered after Sundays and Holidays when the equipment may not have been in operation. This extra load has to be provided for either by considerable excess capacity in the air conditioning equipment or else by providing a considerable longer period of operation before occupancy where such operation is practicable. Usually a compromise is made by having extra capacity for purposes of cooling down the building prior to occupancy and a somewhat longer period of operation after Sundays and Holidays.

There are occasionally installations where the guarantees can only be met by careful adjustment of the equipment. I have one such large installation in mind in which the amount of air handled in the building was restricted by architectural

limitations. In this building it was necessary to shift the cooling load with the travel of the sun. Also, under the specifications, it was required to vary the building temperature between 72 F and 80 F with the variation in outside temperature. As the temperature rose, it was a simple matter to permit a rise of indoor temperature, but where the temperature outside dropped suddenly from one day to the next, it was almost impossible to get a corresponding reduction in inside temperature because it became necessary to cool the entire building structure several degrees. To do this required either an excessive plant capacity—a plant capacity much greater than the largest instantaneous load—or else to operate the plant practically continuously during the period of the drop in temperature. It did not prove practicable to adhere strictly to this schedule of temperatures. Excess capacity was available and employed and additional time of operation was also required in lowering the building temperature to a smaller range than that specified.

Balance between the cooling load provided by the air and the capacity of the refrigerating machines is important in any installation. A margin in the capacity of the refrigeration machine is of little value if there is not adequate air handling capacity to utilize it. In general, there should be an excess of air handling capacity available so that the refrigerating capacity may always be used to the maximum even at the expense of an increase in dewpoint temperature.

Caution should be given against broad conclusions or too great generalizations from a paper such as this. The data as far as external heat load and external heat load and sunlight calculations is concerned undoubtedly applies well to localities corresponding to Detroit, but that would not apply to localities in the southern states . . . as in Texas, for example. The relation between the average load and the maximum load in the two localities are quite different. In Texas the maximum load might be calculated to be approximately the same as in Detroit, while the average load for 24 hours in Texas would be much greater so that the Detroit plant if located in Texas might not have had an excessive operating margin. This points out a defect in our method of calculating heat loads, as already pointed out in this discussion.

There is another point about moisture that is raised, and I would like to confirm the remarks of Mr. Chester. I know in certain types of buildings with air-washer equipment the dewpoint of the building follows very closely the dewpoint of the apparatus. On the other hand, we have found in humidifying in cotton mills that there is a big lag in humidity due to the absorption of moisture by the cotton. The moisture absorption of materials in certain types of construction may also make a great difference. It tends to give a lag and at the same time it tends to smooth out any variation in humidity. Possibly in these buildings the porous ceilings may have had considerable adsorptive power and have smoothed out what would otherwise be irregularities in your observed dewpoints. I don't believe we can make a general assumption regarding humidity control based on the results of these tests, although they are no doubt absolutely correct, because I know there are other conditions where you do get a fairly rapid response to change in apparatus dewpoint.

F. E. Giesecke, College Station, Tex.: The existence of time-lag in the cooling load resulting from electric lighting is new to me and of considerable interest. Evidently, the explanation is that the electric lighting units function like panel warming units and emit a considerable portion of their heat by radiation; the radiated heat is absorbed by the furniture, floor, walls, and ceiling, and then released gradually to the cooling system.

L. T. Avery, Cleveland, Ohio: Panel storage of heat reminds me of a job at the Federal Reserve Bank in Cleveland. It was a similar distribution system of air, where the air was injected through a plenum above the ceiling of the room with a very high Mazda lamp load, and the difficulty came when we tried to turn the system off at night. The refrigerating machine could not be turned off at night. There was sufficient stored heat in the ceiling panels from those lights and from

some sunlight effect, although the sun effect was very small, compared with other loads, that unless the system was run straight through the night it would never pick the load up in the morning. It was entirely adequate to handle the load and loaf along below maximum capacity if permitted to run overnight, but if it were shut off at night it would never get hold of the load before noon the following day. There must have been storage of electric heat in the ceilings.

AUTHOR'S CLOSURE: The points mentioned by Mr. Chester are pertinent. I would like to repeat, however, that the main object of this test was to analyze loads encountered when operating in the usual manner, and in this building the usual manner involved an increase in indoor temperature as the outdoor temperature rises. This method has much in its favor, and is no doubt used in many other buildings.

Mr. Chester is correct in his reference to the effect of changing thermostat settings of July 7; however, this did not apply to the data set forth for the morning hours of August 2, 3 and 4. The building was thus tested in both ways,—with a rising indoor temperature and with a stationary indoor temperature.

As to the effect of the control of chilled water temperature, it was not intended that the discussion in the paper on this point was applicable to buildings generally. The paper deals with a specific building. The author does not entirely agree with Mr. Chester but that point is incidental to the subject of the paper.

Referring to Mr. Hattis' remarks, glass brick was not used primarily to give more natural light near the outside walls. It was chosen because it has the insulating value of double sash while having almost no air infiltration. It also eliminates window washing and window maintenance generally. In the five years since the building has been built the glass bricks have not been washed.

The cooling load caused by the glass brick is less than that caused by an *equal area* of window unless the window has exterior awnings. We do not know what area of window would be *equivalent* from a lighting standpoint.

We did not use water on the roof. While that scheme was considered at the time the building was designed, it was felt that since the roof load was a small part of the total and the mechanical difficulties of maintaining a pool of water on a roof are certainly not negligible, such an installation could not be justified. Instead, the roof was insulated with two inches of cork board.

Replacement of air required for the toilet exhaust system caused about 0.70 building volume change per hour and just about satisfies the minimum fresh air requirement at all times. Aside from this, the suggested operation on all recirculated air at night has always been a part of our operating practice. Generally all refrigeration is cut off at the time building use ceases, and is started again shortly in advance of reoccupation next day. During certain sustained periods of extremely hot weather some refrigeration has been necessary all night.

Mr. Walker is one of the author's business associates and his natural modesty does not let it appear in his comments that it was largely through his efforts that this test work was made possible. His comments on the results of the work are indeed generous; they also serve to underline what the author believes to be the true significance of the findings of the test.

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SPRAY NOZZLE PERFORMANCE IN A COOLING TOWER

By L. M. K. Boelter * and S. Hori,** Berkeley, Calif.

This paper is the second of a series on spray tower performance. The research program is sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of California.

Introduction

The performance of a spray tower depends on the amount of water in the tower at a given time, the degree of dispersion of the water into drops and/or sheets and the velocity of the individual liquid particles. In a previous paper (1) experimental results are reported in which the liquid rate was changed by increasing the pressure on a fixed number of nozzles. This procedure increased the number of drops, decreased their size, increased their initial velocity and changed their trajectory as the nozzle pressure was augmented. In this paper the results of tests are reported in which the liquid rate in a tower of given dimensions was increased both by increasing the nozzle pressure and also by increasing the number of nozzles.

The effect of tower length (equivalent to tower volume for a given cross-section) on the energy transfer from the water to the air was evaluated experimentally for a given number of nozzles of one type. In addition two widely different types of nozzles were compared and finally the effect of two screen plates placed below the nozzles was studied and the effect of two wire meshes compared.

The data and results of the tests are presented without the support of analytical treatment. The writers hope that this information will be part of a body of information which will eventually yield satisfactorily to a generalized algebraic analysis.

Description of the Apparatus

The tower employed for these runs has been described in a previous paper (1). A photograph and a vertical section are included as Figs. 1 and 2 respectively. For these runs the contra-flow side only was employed. The tower was thoroughly insulated so that the heat transfer to the surroundings was negligible.

The nozzles shown in Figs. 3A and 3B which were used to obtain the experimental data presented herein are classified as hollow-cone nozzles (2). Fundamentally, the water enters a whirl chamber (Fig. 3A) or through a fixed curved passage (Fig. 3B) and acquires a rapid rotation. The fluid on exit forms a hollow conical sheet and then breaks up into drops. This type of nozzle, as its name implies, gives a cone shaped spray with the large fraction of the water spraying outwards, away from the center. Although this distribu-

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Numerals in parentheses refer to bibliography.

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tion of the water is undesirable in most cases, much smaller drops result from a hollow-cone nozzle than a solid-cone nozzle of the same capacity (1). The experimental technique utilized was similar to that employed by the earlier investigators who used this tower.

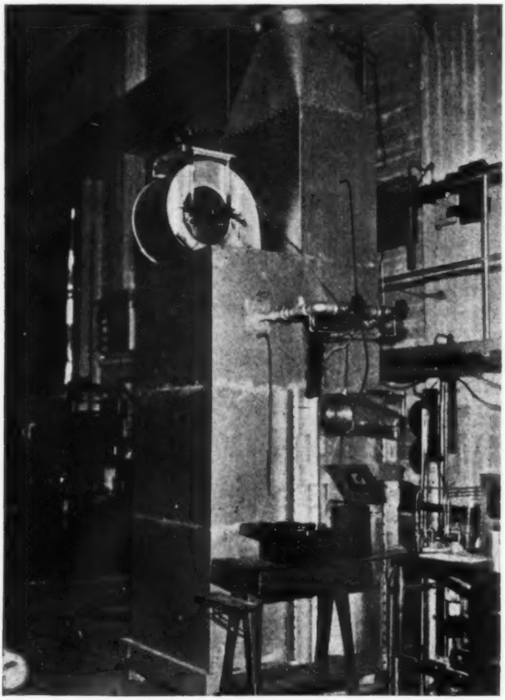
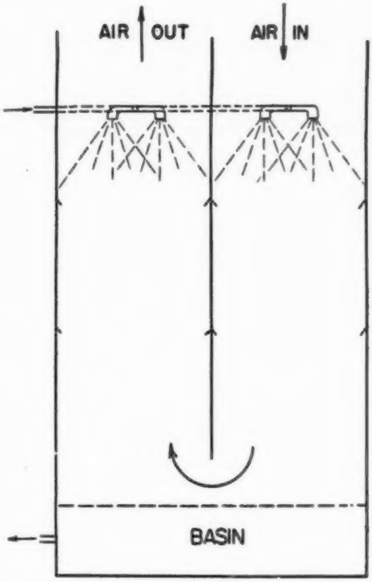


FIG. 1. VIEW OF SPRAY COOLING TOWER SHOWING THERMOCOUPLES, POTENTIOMETER, AND AIR MEASURING DEVICES

UNIT VOLUME ENTHALPY CONDUCTANCE

The results are presented here on the basis of the unit volume enthalpy conductance the concept of which was presented in a previous paper (3). Consideration was given to the range of operation for which the concept is applicable (4). Although practically all data on mass transfer systems are presented on the basis of the unit volume mass conductance, the results are presented here on the basis of the unit volume enthalpy conductance since the cooling towers are essentially energy exchangers and the actual performance is rated on the amount of cooling obtained. Also, the conductance was computed on the basis of the heat sensibly removed from the water, $L_1 \Delta T$, where L_1 is the



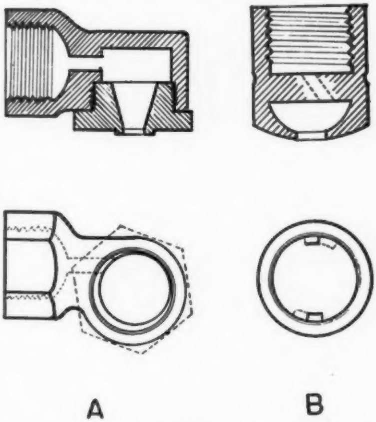


FIG. 3. CROSS-SECTIONS OF TYPICAL HOLLOW-CONE PRESSURE SPRAY NOZZLES
(*A*) Diameter of pipe connection: 3/8 in.
Diameter of discharge orifice: 7/32 in.
(*B*) Diameter of pipe connection: 1/4 in.
Diameter of discharge orifice: 3/16 in.

FIG. 2. (*left*) FLOW DIAGRAM OF SPRAY COOLING TOWER, 4 FT X 3 FT X 7 FT

water rate on to the tower and ΔT is the decrease in temperature of the water, instead of the energy gained by the air in passing through the tower. The performance curves are to be used primarily in the design of cooling towers. The energy increase of the air is practically equal to $L_1 \Delta T$ and the latter will be utilized in this paper.

This unit volume enthalpy conductance is then defined as

$$f_h a = \frac{L_1 \Delta T}{V\,(\Delta h)_{lm}} \quad \text{. . . . . (1a}$$

where $f_h a$, the unit volume enthalpy conductance is the rate of sensible energy transfer per unit of spray volume for a unit of enthalpy potential. This conductance is based on a concept similar to that employed in correlating the performance of spray ponds (5). Its more correct definition for the cases in which the logarithmic mean is applicable (6) is

$$G\,(h_1 - h_2) = (f_h a)\; V\,(\Delta h)_{lm} \quad \text{. . . . . (1b)}$$

The unit volume enthalpy conductance established for this tower is made up of two contributions, (*a*) due to the drops, (*b*) due to the liquid sheet flowing down the tower walls. Roughly these unit conductances are additive for the systems operate in parallel. The sheet cooling apparently contributes about one-fourth of the total unit conductance (1). In the Discussion of Results no attempt is made to separate the wall (liquid sheet) contribution from the total cooling. Generalization of these results in a tower of a

TABLE 1—TABULATION OF DATA AND RESULTS FOR TESTS USING VARIOUS NUMBERS OF NOZZLES A

Run No.	Number of Nozzles	Pressure at Nozzles	Nozzle Discharge Rate	Water Conditions: Inlet Temp	Water Conditions: Outlet Temp	Water Conditions: Sensible Heat Loss	Psychrometric Properties of Air: Rate	Inlet Air: Dry-bulb Temp	Inlet Air: Wet-bulb Temp	Inlet Air: Relative Humidity	Outlet Air: Dry-bulb Temp	Outlet Air: Wet-bulb Temp	Outlet Air: Relative Humidity	Mass Balance: Water Vapor gained by Air	Mass Balance: Measured Rate of Evaporation	Mass Balance: Diff	Energy Balance: Energy Gained by Air	Energy Balance: Total Energy loss by Water	Energy Balance: Diff	Log. Mean Enthalpy Potential	Unit Volume Enthalpy Conductance
	N	P	L_1	T_1	T_2	$L_1\Delta T$	G	t_{2DB}	t_{2WB}	R_2	t_{1DB}	t_{1WB}	R_1	$G\Delta H$	ΔL	D_L	$G\Delta h$	$L_1\Delta T + \Delta L_1 h_L$	D_h	$(\Delta h)_{lm}$	$f_h a$
		psig	lbs. per hr.	°F	°F	Btu per lb	lbs. per hr.	°F	°F	%	°F	°F	%	lbs. per hr.	lbs. per hr.	%	Btu per hr.	Btu per hr.	%	Btu per lb d.a.	Btu per hr ft³ (Btu/lb)
17c	6	35	7700	90.2	80.1	77,800	4570	66.4	60.8	72.3	79.0	79.0	100	52.5	60.8	-12.63	71,400	81,420	-13.30	15.15	146.7
18c		30	7150	90.0	79.1	77,900	4560	65.5	58.1	63.9	77.0	77.0		52.3	61.2	-14.61	70,000	80,780	-8.89	16.53	134.5
20c		25	6460	90.1	77.3	82,700	4560	63.4	51.1	41.3	74.1	74.1		59.4	66.1	-10.12	77,100	85,700	-10.02	19.22	122.9
19c		20	5780	90.1	78.3	68,150	4540	66.2	57.5	58.5	74.7	74.7		47.5	54.4	-12.69	61,500	79,670	-13.26	17.37	112.1
21c		15	4960	90.1	76.4	67,500	4650	65.9	52.5	39.0	71.6	71.6		52.8	56.8	-7.04	64,100	79,020	-8.45	19.65	98.1
22c		10	3965	90.0	75.9	56,000	4710	68.0	53.9	38.2	71.3	69.9	93.2	45.9	48.6	-5.55	54,000	58,140	-7.11	19.31	82.9
23c		5	2535	90.0	74.8	39,500	4650	69.8	54.0	33.6	70.7	66.3	76.5	33.0	36.2	-8.84	39,200	40,010	-2.02	19.94	55.1
14c	5	30	6080	90.7	79.9	65,600	4600	67.3	61.9	73.6	77.7	77.7	100	45.85	50.9	-9.92	61,900	69,040	-9.02	15.05	119.9
15c		20	4920	90.9	79.3	57,100	4590	67.7	62.3	73.9	76.5	76.5		41.24	45.6	-9.45	50,350	59,260	-15.03	15.49	105.3
16c		10	3365	90.6	78.1	42,100	4570	69.2	62.5	68.7	73.7	73.7		33.39	41.4	-19.30	41,450	44,010	-5.81	16.26	74.0
10c	4	30	4870	91.8	79.2	61,400	4450	72.7	61.9	53.9	77.5	77.5	100	49.15	54.3	-9.44	59,100	63,960	-7.76	16.19	108.3
11c		30	4920	90.8	78.6	60,000	4690	68.3	52.4	72.0	76.8	76.8		41.40	45.9	-9.80	54,900	62,140	-11.48	15.4	111.3
12c		20	3960	90.9	78.4	49,500	4530	68.7	63.4	74.5	75.7	75.7		36.10	38.4	-5.99	47,750	51,280	-6.88	15.4	91.9
13c		10	2677	90.7	77.3	35,820	4590	69.2	63.0	70.9	72.6	72.6		29.28	29.8	-1.68	37,000	37,140	-0.46	16.0	64.0
4c	3	30	3635	90.2	75.8	52,300	4800	70.5	60.1	54.1	72.9	72.9	100	42.0	45.15	-6.97	49,100	54,280	-9.55	16.09	93.0
5c		20	2918	90.0	75.5	42,500	4710	70.0	60.1	57.4	71.9	71.3	97.1	35.05	36.05	-2.77	41,100	44,070	-6.74	16.21	74.9
6c		10	1991	90.2	77.4	25,600	4580	76.8	66.0	56.1	75.4	72.6	87.4	24.77	25.6	-3.24	25,470	26,660	-4.46	14.50	50.3
1c	2	30	2422	90.7	75.4	37,150	4690	72.4	62.0	55.3	72.7	71.5	94.4	32.0	35.0	-8.56	35,370	38,760	-8.53	15.69	67.9
2c		20	1951	90.2	75.8	27,900	4690	74.7	64.0	56.3	72.8	70.6	85.1	25.0	29.5	-6.34	24,950	28,190	-4.68	15.92	52.1
3c		10	1309	89.8	72.5	22,610	4730	72.8	58.8	42.4	70.7	66.3	79.4	25.8	24.0	+4.9	26,800	23,590	+4.96	16.62	38.8
7c	1	30	1267	91.3	73.7	22,900	4660	75.2	63.7	52.7	72.2	69.7	88.1	23.5	25.2	-6.64	22,220	23,350	-4.84	14.69	43.4
8c		20	987	89.9	73.1	16,590	4640	75.1	62.5	48.7	71.9	67.9	84.5	21.08	21.0	+0.33	19,490	17,460	+11.69	14.99	31.6
9c		10	651	90.2	72.8	14,330	4610	75.5	62.9	49.9	73.3	67.3	73.2	17.89	16.0	+18.55	14,670	11,990	+22.36	14.60	22.2

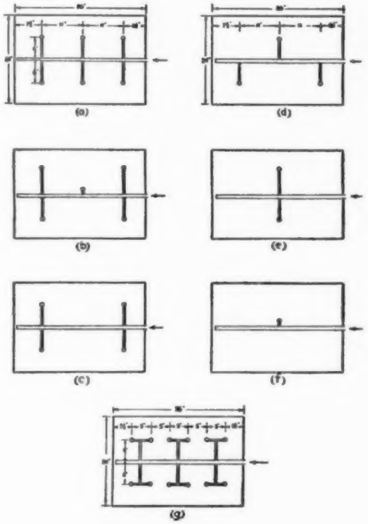
different cross-section, employing similar nozzles, can be accomplished by utilizing the liquid rate per unit cross-sectional area of the tower (L_1/S) and calculating the corresponding unit volume conductances. This procedure will not yield the correct result because of the contribution of the wall sheets to energy transfer (*i.e.*, cooling), but it is proposed as a working approximation.

DISCUSSION OF THE RESULTS

Effect of Number of Nozzles in Tower: The tests of the first series were performed to investigate the over-all performance of the spray cooling tower

FIG. 4. CROSS-SECTION OF SPRAY CHAMBER SHOWING POSITIONS OF NOZZLES FOR TESTS USING VARIOUS NUMBERS OF NOZZLES

1 in. nominal size main pipe
(*a*) to (*f*) Nozzles *A* ⅜ in. nominal size pipe arms
(*g*) Nozzles *B* ¼ in. nominal size pipe arms



using various numbers of spray nozzles in the spray chamber as shown in Fig. 4. Fig. 3A illustrates the nozzles employed in the spray chamber for these runs. These runs were performed with a constant inlet water temperature, constant tower volume, constant gas rate and varying water rates. The data and results are tabulated in Table 1. Fig. 5 presents a series of curves showing the isothermal discharge rates of the nozzles as a function of the pressure at the nozzle and the number of nozzles. The discharge rates of the nozzles are nearly proportional to the square root of the pressure difference, as stated by H. G. Houghton (2).

In Fig. 6 are presented the results of the first series of tests. Here, the unit volume enthalpy conductance is plotted as a function of the pressure at the nozzles with the number of nozzles in the spray chamber as the parameter. A series of constant water rate (nozzle discharge) lines are also plotted. From this graph, the effect of the number of nozzles and the pressure at the nozzles for a given water rate is evident. For a given water rate, a comparatively

slight increase in conductance is obtained by reducing the number of nozzles, while a large increase in nozzle pressure is required to maintain the same water rate. This effect is more noticeable at the lower water rates. These results may be explained by a consideration of the performance of the nozzles under the various conditions.

It is known that for pressure nozzles, the average drop size is approximately inversely proportional to the square root of the pressure difference, that is,

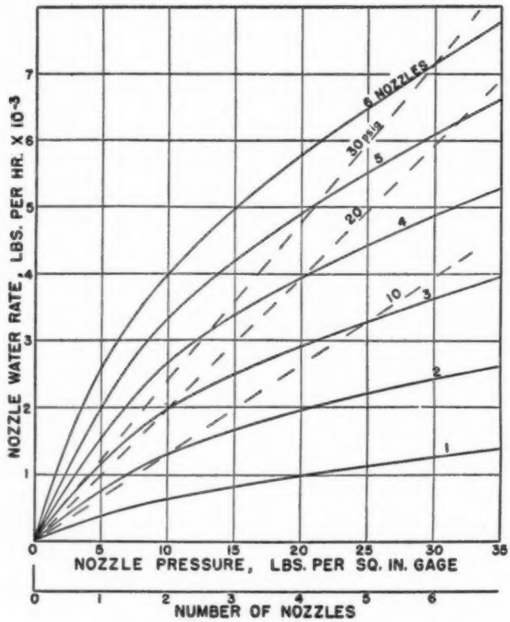


FIG. 5. DISCHARGE CHARACTERISTICS OF THE NOZZLES (A)

Legend:

——— Nozzle pressure as abscissa
– – – – Number of nozzles as abscissa
$L_1 = 181NP^{0.556}$ expresses the results for inlet water temperature $T = 90$ F

inversely proportional to the liquid rates for a given liquid (2). Therefore, for a given water rate and type of nozzle, as the number of nozzles is increased, the required pressure is reduced, thus producing larger drops. Also, the greater the number of nozzles in a given tower volume, the greater the chance of interference between the various sprays. At the higher pressures, the drops will approach a limiting size which will depend on the type of nozzle and the dynamical conditions in the tower. This effect will partially account for the flattening of the curves at the higher pressures.

The empirical equations noted in Fig. 6 reveal that the exponent of the liquid rate term decreases with an increase in the number of nozzles. This means that the increase in the conductance is less for a given increase in the liquid rate for a large number of nozzles, as contrasted with a few. The result can be partially explained by the fact that the effect of interference between the various sprays is increased in the case of the greater number of nozzles and the greater quantity of water within the spray chamber at any given time will

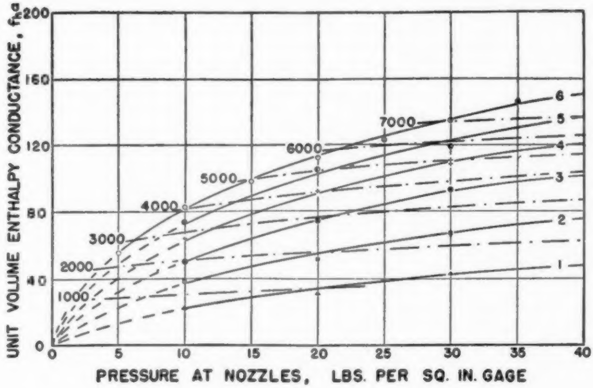


FIG. 6. UNIT VOLUME ENTHALPY CONDUCTANCE *vs.* PRESSURE AT NOZZLES FOR VARIOUS NUMBERS OF NOZZLES IN SPRAY CHAMBER

Operating Conditions:
Inlet water temp. = 90 F.
Air rate = 4600 lb of dry air per hour
Nozzles (A)
Tower volume = 35 ft^3
Legend:
——— Fixed number of nozzles
–·–·–·– Constant water rate at nozzles, lb per hour

Unit volume enthalpy conductance as a function of water rate on the tower for 1 to 6 nozzles:

$$(f_h a)_1 = 0.288 L_1^{1.03}$$
$$(f_h a)_2 = 0.0244 L_1^{1.02}$$
$$(f_h a)_3 = 0.0342 L_1^{0.965}$$
$$(f_h a)_4 = 0.0476 L_1^{0.914}$$
$$(f_h a)_5 = 0.0716 L_1^{0.856}$$
$$(f_h a)_6 = 0.096 L_1^{0.816}$$

decrease the magnitude of the drop surface area per unit volume (a) and may decrease the value of the driving potential.

Fig. 7 is a cross-plot of Fig. 6 and reveals the unit volume enthalpy conductance as a function of the number of nozzles for various pressures at the nozzles. Constant water on tower rate lines are also shown on this plot. It would be expected that if there were no interference between the sprays, the unit volume enthalpy conductance would increase linearly with the number of nozzles. This effect is clearly seen by considering the experimental points shown on the graph. For the 30 and 20 lb per square inch pressures at the nozzles, the first three points corresponding to 1, 2, and 3 nozzles, lie nearly along a straight line and the curve gradually deviates from a line drawn through these points. The decrease in slope is due to the interference between the various sprays. At a pressure of 10 lb per square inch gage, the first four points lie approximately along a straight line. Again, the result may be attributed to the fact that there is less interference at the low pressure due to

the smaller cone angle of the spray. From a consideration of the constant water lines, it can be stated that the relative cooling advantage of using less nozzles and a higher pressure is greater at the higher water rates.

Effect of Length of Spray Chamber: Another series of runs was made to investigate the effect of the length of the tower on the over-all performance of the system. These runs were made with the type A nozzles at three different

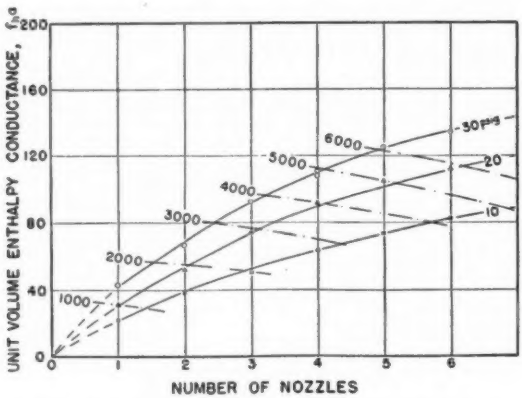


FIG. 7. UNIT VOLUME ENTHALPY CONDUCTANCE *vs.* NUMBER OF NOZZLES FOR VARIOUS PRESSURES AT NOZZLES

Operating Conditions:
Inlet water temperature = 90 F
Air rate = 4600 lb of dry air per hour
Nozzles (*A*)
Tower volume = 35 ft³

Legend:
——— Constant pressure at nozzles, psig
–·–·–·– Constant nozzle water rate, lb per hour

heights in the spray chamber (Fig. 8) while maintaining the inlet water temperature and gas rate approximately constant and varying the water rate to the nozzles.

The results in this case were presented on the basis of the total volume enthalpy conductance which is expressed by

$$(f_h a V) = \frac{L_1 \Delta T}{(\Delta h)_{lm}} \quad \text{. . . . . (1c}$$

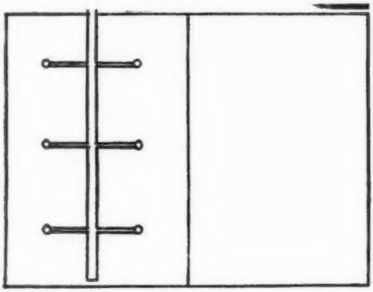
where ($f_h aV$) is the rate of transfer of sensible heat for the total chamber volume per unit of enthalpy potential. This conductance is used so that the relative performances can be compared in terms of values independent of the actual tower volume.

This series of test results are shown in Fig. 9 (also Table 2) where the total volume enthalpy conductances are plotted as a function of the tower volume for various pressures at the nozzles. The curves indicate that the total volume enthalpy conductance approaches a limit as the tower length is increased, if the operating conditions are maintained invariable. The lower the nozzle pressure or water rate, the more quickly the limit is approached. Beyond an optimum

tower length it would be uneconomical to further increase the length of the tower to obtain the correspondingly slight increase in the conductance. This rapid approach to a maximum conductance substantiates the conclusion (1) that the greater portion of the cooling in the tower takes place near the entrance of the water into the tower. Therefore, at the lower pressures, since the cone angle is small, a large portion of the water travels down the center of the tower and is cooled rapidly and quickly approaches the practical limit to which the water can be cooled. Measurements by H. F. Johnstone and R. V. Kleinschmidt (8) designed to determine absorption coefficients in spray towers also indicated that the rate of absorption was high immediately in front of the nozzles and that it dropped off rapidly with distance.

The rapid cooling of the drops near the nozzles may be attributed to the high velocities of the drops on discharging from the orifice and the high potential at the liquid entrance end of the tower. The high velocities tend to decrease the resistance to mass transfer on the gaseous side of the liquid-gas interface, thus resulting in high rates of evaporation and heat transfer. Due to the small mass of the drops and the resistance of the counterflowing air, the drops are quickly decelerated to their limiting velocity (9) and slowly drift downward.

The total volume enthalpy conductances as a function of the pressure at the nozzles for the three volumes considered are plotted in Fig. 10. A comparison of the values of the exponent of the empirical equations noted on the figure demonstrates that within the normal operating range, the rate of in-



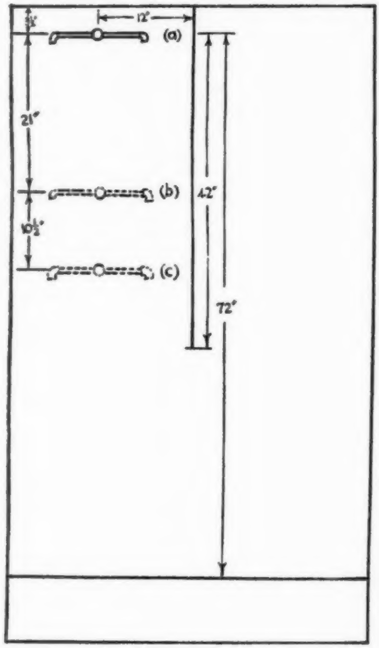


FIG. 8. ELEVATION SECTION OF SPRAY CHAMBER SHOWING ELEVATIONS OF SPRAY NOZZLES ABOVE TOWER OUTLET

Legend:
(a) Tower volume = 35 cu ft
(b) Tower volume = 24.5 cu ft
(c) Tower volume = 19.3 cu ft

TABLE 2—TABULATION OF DATA AND RESULTS OBTAINED FOR VARIOUS HEIGHTS OF THE SPRAY CHAMBER

Run No	Pressure at Nozzles	Nozzle Discharge Rate	Water Conditions			Psychrometric Properties of Air							Mass Balance			Energy Balance			Log Mean Enthalpy Potential	Unit Volume Enthalpy Conductance	Total Volume Enthalpy Conductance
			Inlet Temp	Outlet Temp	Sensible Heat Loss	Rate	Inlet Air			Outlet Air			Water Vapor gained by Air	Measured Rate of Evaporation	Diff.	Energy gained by air	Total Energy Loss by water	Diff.			
							Dry-bulb Temp	Wet-bulb Temp	Relative Humidity	Dry-bulb Temp	Wet-bulb Temp	Relative Humidity									
	P	L_1	T_1	T_2	$L_1\Delta T$	G	t_{2DB}	t_{2WB}	R_2	t_{1DB}	t_{1WB}	R_1	$G\Delta H$	ΔL	D_L	$G\Delta h$	$L_1T_1 - L_2h_2$	D_h	$(\Delta h)_m$	f_ha	f_haV
	psig	lbs. per hr.	°F	°F	Btu per hr.	lbs. d.a. per hr.	°F	°F	%	°F	°F	%	lbs. per hr.	lbs. per hr.	%	Btu per hr.	Btu per hr.	%	Btu per lb. d.a.	Btu per hr. ft.³ (Btu/lb.)	Btu per hr. (Btu/lb.)
TOWER VOLUME = 35 cu. ft.																					
17c	35	7700	90.2	80.1	77,800	4570	66.4	60.8	72.3	79.0	79.0	100	52.5	60.8	-12.63	71,400	81,420	-12.30	15.15	146.7	5130
18c	30	7150	90.0	79.1	77,900	4560	65.5	58.1	63.9	77.0	77.0		52.3	61.2	-14.61	70,000	80,780	-8.89	16.53	134.5	4700
20c	25	6460	90.1	77.3	82,700	4560	63.4	51.1	41.3	74.1	74.1		59.4	66.1	-10.12	77,100	85,700	-10.02	19.22	122.9	4300
19c	20	5780	90.1	78.3	68,150	4540	66.2	57.5	58.5	74.7	74.7		47.5	54.4	-12.69	61,300	70,670	-13.26	17.37	112.1	3820
21c	15	4960	90.1	76.4	67,500	4650	65.9	52.5	39.0	71.6	71.6		52.8	56.8	-7.04	64,100	70,020	-8.45	19.35	98.1	3437
22c	10	3965	90.0	75.9	56,000	4710	68.0	53.9	38.2	71.3	69.9	93.2	45.9	48.6	-5.55	54,000	58,140	-7.11	19.31	82.9	2900
23c	5	2535	90.0	74.8	38,500	4650	69.8	54.0	33.6	70.7	66.3	76.5	33.0	36.2	-8.84	39,200	40,000	-2.02	19.94	55.1	1930
TOWER VOLUME = 24.5 cu. ft.																					
1e	35	7650	90.0	80.1	75,700	4500	69.2	59.5	56.1	77.7	77.7	100	53.7	62.4	-13.93	68,200	78,700	-13.31	16.09	192.0	4710
2e	30	7110	90.1	79.7	74,000	4530	70.2	58.3	48.2	76.8	76.8		55.5	61.5	-9.76	68,100	76,940	-11.50	16.97	178.0	4360
3e	25	6460	90.0	79.4	68,500	4510	71.3	58.3	44.9	75.7	75.7		53.1	59.5	-10.76	63,000	71,200	-11.59	16.98	164.8	4030
4e	20	5800	90.0	79.3	62,050	4500	73.9	59.5	41.6	74.9	74.9		50.3	54.9	-8.39	56,000	64,610	-13.31	17.08	148.2	3640
5e	15	5040	90.0	78.5	58,000	4540	68.3	58.3	54.5	73.2	73.2		43.1	48.5	-11.12	52,550	60,260	-12.80	17.69	134.0	3280
6e	10	4075	90.0	78.6	46,500	4490	72.6	59.2	44.3	73.3	71.6	92.3	38.9	43.3	-10.17	42,760	48,520	-15.85	18.03	105.1	2579
7e	7.5	3540	90.0	78.2	41,750	4560	72.2	58.4	42.5	72.5	70.1	88.7	36.4	45.9	-20.7	40,000	43,870	-8.83	18.92	90.0	2204
8e	5	2740	90.0	74.4	42,740	4560	57.4	45.2	36.0	62.6	60.3	87.5	31.9	35.6	-10.4	40,400	44,250	-8.70	24.59	71.0	1740
9e	3	1770	90.0	76.1	24,600	4480	61.2	46.1	27.5	62.8	56.2	65.3	21.5	22.5	-4.26	25,050	25,590	-1.99	26.40	38.0	932
TOWER VOLUME = 19.3 cu. ft.																					
1f	35	7690	90.0	82.1	60,700	4490	73.0	64.0	60.9	78.7	78.7	100	47.8	47.3	+1.05	58,600	63,070	-7.09	15.63	201.0	3880
2f	30	7160	90.1	81.3	63,050	4580	71.5	61.9	57.8	77.1	77.1		48.2	55.2	-12.65	58,800	65,770	-10.6	16.31	200.2	3860
3f	25	6520	90.0	81.6	54,800	4510	73.6	63.7	57.8	77.3	77.3		44.9	46.3	-3.02	53,300	57,100	-6.65	15.63	181.5	3500
4f	20	5840	90.0	81.9	47,300	4510	75.3	66.6	63.1	77.6	77.6		38.1	41.5	-8.2	44,600	48,970	-8.67	14.60	167.9	3240
5f	15	5095	90.0	81.9	41,250	4450	77.0	67.4	60.5	77.4	77.0		34.6	37.6	-7.98	39,600	43,130	-10.5	14.69	145.7	2810
6f	10	4120	90.0	77.5	51,500	4520	60.8	53.5	61.8	68.6	68.6		36.1	40.9	-11.73	47,900	53,360	-10.23	20.92	127.6	2460
7f	7.5	3560	90.0	77.7	43,800	4500	63.0	54.9	60.3	68.5	68.0	97.5	32.0	36.0	-11.11	41,700	45,450	-8.25	20.70	109.6	2118
8f	5	2800	90.0	78.9	31,100	4500	69.9	59.2	52.6	70.7	68.4	89.0	27.4	25.8	+6.2	29,900	32,310	-4.37	19.61	82.6	1592

crease of the total volume enthalpy conductance with liquid rate is higher for a longer tower volume than for a shorter tower. As the pressure on the nozzles is increased the average drop size is decreased and for longer towers the coalesced small drops (corresponding to the higher nozzle pressure) will tend to increase the term (a) in the product ($f_h a$) over that which obtains in the longer tower operating at the lower nozzle pressure. Increased tower length increases both (f_h) and (aQ in the product ($f_h a$), the former because of the

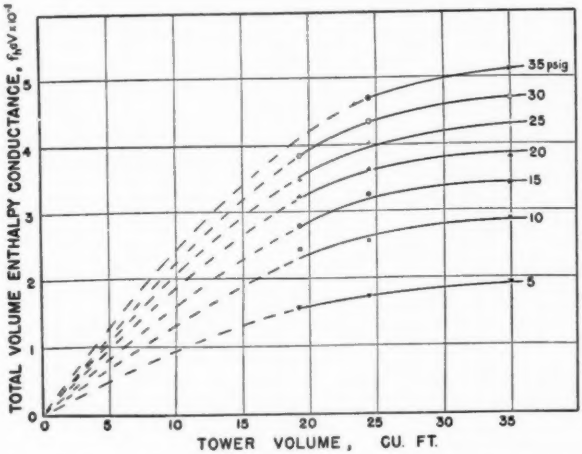


FIG. 9. TOTAL VOLUME ENTHALPY CONDUCTANCE *vs.* TOWER VOLUME FOR VARIOUS PRESSURES AT NOZZLES

Operating Conditions:
Inlet water temp. = 90 F
Air rate = 4600 lb of dry air per hour
Six nozzles *Type A*

effect on the average velocity, the latter because there are more drops in the tower.

The curves shown in Fig. 11 are the unit volume enthalpy conductance plotted as a function of the tower volume for various pressures at the nozzles. The value of the exponent in the equations noted below Fig. 9 compare favorably with the results obtained by A. W. Hixson and C. E. Scott for the variation of the over-all absorption coefficients, $K_g a$, for ammonia and sulfur dioxide by liquid water and benzene vapor by straw oil. For a constant liquid rate and gas rate, the absorption coefficient for all three cases was found to vary approximately with the -0.5 (average of a series of experimental curves) power of the tower length.

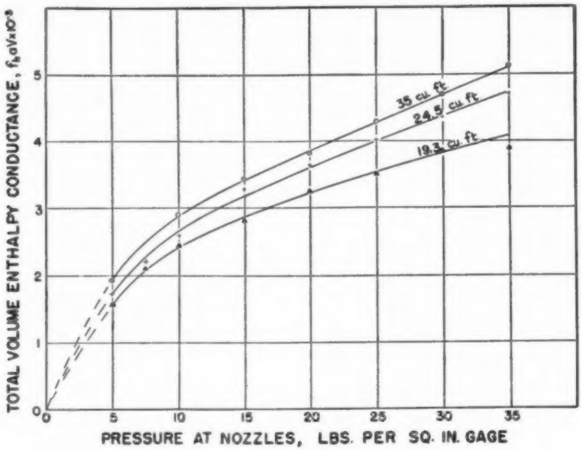


Fig. 10. Total Volume Enthalpy Conductance *vs.* Pressure at Nozzles for Various Tower Volumes

Operating Conditions:
Inlet water temp. = 90 F
Air rate = 4600 lb of dry air per hour
Six nozzles, *Type A*

Corresponding equations in terms of liquid rate:
$(f_h aV)_{V=35} = 3.36L^{0.816}$
$(f_h aV)_{V=24.5} = 3.88L^{0.79}$
$(f_h aV)_{V=19.3} = 4.83L^{0.75}$

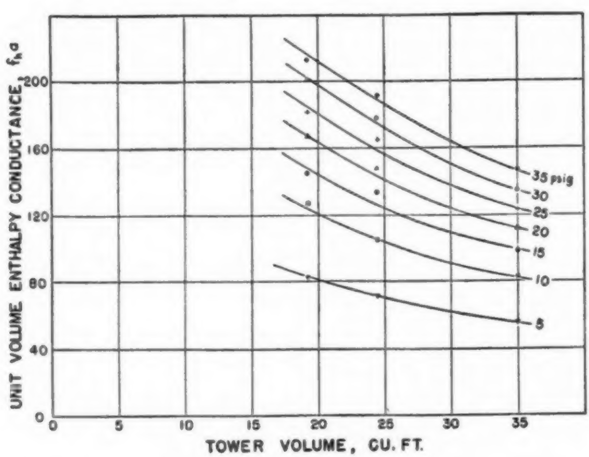


Fig. 11. Unit Volume Enthalpy Conductance *vs.* Tower Volume for Various Pressures at Nozzles

Operating Conditions:
Inlet water temp. = 90 F
Air rate = 4600 lb of dry air per hour
Six nozzles *Type A*

The corresponding equations for three nozzle pressures:
$(f_h a)_{30} = 1557\,V^{-0.[illegible]}$
$(f_h a)_{20} = 1245\,V^{-0.[illegible]}$
$(f_h a)_{10} = 919\,V^{-0.[illegible]}$

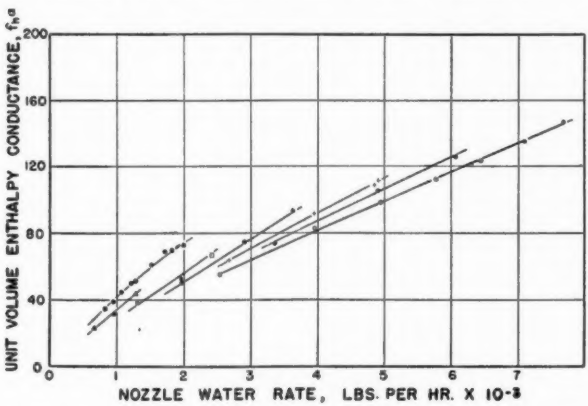


FIG. 12. UNIT VOLUME ENTHALPY CONDUCTANCE *vs.* NOZZLE WATER RATE FOR VARIOUS NUMBERS OF NOZZLES IN THE SPRAY CHAMBER

Legend:
12 nozzles *Type B* Inlet water temp. = 90 F
6 nozzles *Type A* Dry air rate = 4600 lb per hour

The relationships obtained by Hixson and Scott were:

$$K_g a = \frac{C \cdot G^{0.8} \cdot L}{H^{0.5}}$$

(Ammonia by water)

$$= \frac{C' \cdot G^{0.8} \cdot L^{0.9}}{H^{0.5}}$$

(Sulfur dioxide by water)

$$= \frac{C'' \cdot G^{0.8} \cdot L^{0.9}}{H^{0.5}}$$

(Benzene vapor by straw oil)

Within the range of operation of the tower the unit volume mass conductance ($K_g a$) and the unit volume enthalpy conductance ($f_h a$), differ little (4). Therefore, it is justifiable to compare the various relationships obtained for $f_h a$ with the expressions of $K_g a$ of other investigators (10, 12, 13) when the mass transfer systems are similar. In all the systems being considered, the liquid side resistance at the liquid-gas interface is almost negligible as compared to the gas side resistance.

The values obtained for the exponent of the liquid rate ranged from 0.816 to 1.03 which compare very well with the mean values of 1.0 and 0.9 obtained by Hixson and Scott.

Effect of Type of Nozzle: Fig. 12 is a plot of the unit volume enthalpy conductance as a function of the water rate for the various numbers of nozzles of *Type A,* and also a curve obtained with 12 nozzles of *Type B* (see Fig. 3*B*) placed in the same spray chamber (Fig. 4*g*). This set of curves clearly shows the greater unit volume enthalpy conductance for the nozzles which generate

TABLE 3—TABULATION OF DATA AND RESULTS OF TESTS USING NOZZLES B

Run No.	Pressure at Nozzles	Nozzle Discharge Rate	Water Conditions			Psychrometric Properties of Air							Mass Balance			Energy Balance			Log Mean Enthalpy Potential	Unit Volume Enthalpy Conductance
			Inlet Temp.	Outlet Temp.	Sensible Heat Loss	Rate	Inlet Air Dry-bulb Temp.	Inlet Air Wet-bulb Temp.	Inlet Air Relative Humidity	Outlet Air Dry-bulb Temp.	Outlet Air Wet-bulb Temp.	Outlet Air Relative Humidity	Water Vapor gained by Air	Measured Rate of Evap.	Diff.	Energy gained by air	Total Energy Loss by Water	Diff.		
	P	L_1	T_1	T_2	$L_1\Delta T$	G	t_{2DB}	t_{2WB}	R_2	t_{1DB}	t_{1WB}	R_1	$G\Delta H$	ΔL	D_L	$G\Delta h$	$L_1 T \Delta h_L$	D_h	$(\Delta h)_{lm}$	$f_h a$
	psig.	lbs per hr.	°F	°F	Btu per lb.	lbs. d.a. per hr	°F	°F	%	°F	°F	%	lbs. per hr.	lbs. per hr.	%	Btu per hr.	Btu per hr	%	Btu per lb d.a.	Btu per hr-(Btu/lb)
1b	47.5	2000	89.8	70.9	37,800	4550	71.2	58.3	45.3	69.6	69.6	100	36.6	42.0	-12.85	38,300	39,430	-2.95	14.87	72.6
2b	39.8	1840	91.5	70.7	38,220	4490	73.0	58.5	40.0	69.9	69.6	98.4	37.7	37.7	0	37,220	39,680	-6.15	15.60	70.0
3b	35.0	1727	91.8	70.0	37,600	4600	68.5	58.4	54.1	69.2	69.2	100	33.2	35.0	-5.15	37,100	38,930	-4.70	15.45	69.5
4b	26.9	1520	89.8	70.1	30,000	4550	71.0	59.7	50.9	69.6	68.5	94.6	29.1	30.85	-5.66	30,100	31,180	-3.46	14.10	60.8
7b	22.0	1290	90.7	68.6	28,530	4490	70.8	56.7	40.0	68.5	67.1	93.1	33.3	33.3	0	33,900	29,750	+3.97	15.43	51.8
5b	18.0	1223	91.6	69.8	26,850	4600	71.0	57.9	44.1	69.1	66.4	93.1	32.2	29.0	+7.65	29,650	27,950	+5.63	15.31	50.1
8b	14.8	1052	90.6	68.0	23,800	4460	72.2	56.6	36.2	68.9	66.9	90.5	33.6	29.85	+12.2	33,150	24,875	+33.3	15.13	44.9
6b	12.0	961	90.5	68.6	21,050	4550	67.9	57.2	51.2	67.6	64.3	83.4	20.8	22.4	-7.15	22,400	21,870	+2.42	15.86	39.0
9b	10.0	835	90.5	68.8	16,110	4380	73.5	57.7	36.8	70.7	67.6	85.3	31.7	22.4	+41.5	31,600	18,935	+66.9	14.84	34.8

TABLE 4—TABULATION OF DATA AND RESULTS OF TESTS PERFORMED WITH SCREEN PACKINGS IN THE SPRAY CHAMBER

Run No.	Pressure at Nozzles	Nozzle Discharge Rate	Water Conditions			Psychrometric Properties of Air							Mass Balance			Energy Balance			Log. Mean Enthalpy Potential	Unit Volume Enthalpy Conductance
			Inlet Temp.	Outlet Temp.	Sensible Heat Loss	Rate	Inlet Air Dry-bulb Temp.	Inlet Air Wet-bulb Temp.	Inlet Air Relative Humidity	Outlet Air Dry-bulb Temp.	Outlet Air Wet-bulb Temp.	Outlet Air Relative Humidity	Water Vapor gained by Air	Measured Rate of Evap.	Diff.	Energy gained by air	Total Energy Loss by Water	Diff.		
	P	L_1	T_1	T_2	$L_1\Delta T$	G	t_{2DB}	t_{2WB}	R_2	t_{1DB}	t_{1WB}	R_1	$G\Delta H$	ΔL	D_L	$G\Delta h$	$L_1 T \Delta h_L$	D_h	$(\Delta h)_{lm}$	$f_h a$
	psig.	lbs per hr.	°F	°F	Btu per lb.	lbs. d.a. per hr	°F	°F	%	°F	°F	%	lbs. per hr.	lbs. per hr.	%	Btu per hr.	Btu per hr.	%	Btu per lb. d.a.	Btu per (hr) (Btu/lb)

TWO-#8 MESH GALVANIZED SCREEN PACKINGS																				
7	35	7680	90.2	81.3	68,350	4490	72.8	66.5	71.7	81.8	81.8	100.0	50.2	56.8	-11.61	65,000	71,150	-8.65	12.1	161.2
8	30	7115	89.9	81.2	61,900	4500	73.7	68.3	75.8	81.3	81.3	100.0	43.5	46.6	-6.66	55,950	64,190	-12.85	11.36	155.7
1	25	6455	89.5	78.8	69,100	4550	70.5	62.4	63.5	78.6	78.1	97.9	47.9	55.4	-13.54	61,500	71,690	-14.20	13.7	144.1
9	24	6390	90.1	78.8	72,250	4575	71.4	62.4	57.0	78.6	78.6	100.0	53.9	56.8	-5.12	66,900	74,905	-10.7	14.55	141.9
2	20	5710	88.8	77.1	63,400	4565	62.8	55.6	63.4	74.9	73.8	94.9	45.5	56.8	-19.90	63,200	65,960	-4.19	16.8	109.1
3	15	4910	90.2	77.4	63,300	4655	70.1	60.6	52.5	75.9	75.7	99.3	47.0	55.4	-15.17	57,750	65,810	-12.25	15.2	119.0
10	12.2	4560	89.6	77.0	57,600	4575	71.4	61.5	56.5	75.1	74.9	99.1	42.3	44.7	-5.38	50,350	59,620	-15.6	14.62	112.6
4	10	3880	90.1	77.4	49,300	4570	69.2	60.1	58.9	74.2	73.8	97.2	39.7	43.7	-17.46	49,700	51,460	-3.46	16.3	86.5
6	7.5	3330	90.1	75.9	47,250	4575	71.9	58.8	44.8	73.3	72.0	94.1	41.5	46.6	-5.04	46,750	49,170	-4.93	16.9	80.0
5	5	2490	90.1	75.1	37,350	4565	73.7	61.5	49.3	72.2	71.4	97.5	35.5	33.5	+5.97	37,300	38,795	-3.85	15.4	69.3
11	3	1630	90.1	74.0	26,200	4615	67.8	60.1	64.1	68.9	66.1	86.5	17.3	16.75	+6.95	20,150	26,900	[illegible]	15.2	49.2
TWO-#4 MESH GALVANIZED SCREEN PACKINGS																				
12	34.9	7675	90.0	78.3	69,750	4515	63.9	57.5	68.5	78.8	78.8	100	57.1	67.0	-14.76	79,250	92,850	-14.65	15.25	168.1
13	30.0	7110	90.5	79.0	81,750	4460	67.8	58.5	56.1	79.0	79.0	↓	59.2	68.6	-14.1	77,000	84,990	-9.40	16.8	139.0
14	25	6515	90.3	79.3	71,600	4480	70.2	62.0	67.7	79.0	79.0	↓	48.3	58.8	-17.9	66,400	74,380	-10.7	14.45	141.6
15	20	5805	89.5	78.8	62,100	4540	70.2	62.7	65.7	77.8	77.8	↓	46.3	53.4	-13.3	59,100	64,600	-8.51	13.0	136.7
16	15	4920	90.4	78.8	57,000	4500	70.1	63.7	70.3	77.3	77.3	↓	41.2	48.05	-14.25	53,900	59,225	-10.7	13.77	118.2
17	10	4020	90.3	77.6	51,000	4590	65.5	60.2	73.0	72.7	72.7	↓	41.7	43.7	-4.58	45,500	52,995	-14.13	17.04	85.5
18	7.5	3430	90.3	76.8	46,300	4565	71.9	61.9	57.5	74.4	74.4	↓	39.6	34.95	+13.30	46,000	47,865	-3.90	15.08	87.9
19	5	2585	90.0	76.2	35,650	4660	75.8	64.7	53.2	75.6	73.9	89.6	31.9	32.8	-2.75	35,100	37,100	-5.39	14.07	72.5
20	3	1585	90.0	75.6	22,800	4600	76.0	62.8	47.0	73.7	67.6	73.0	18.1	20.4	-11.26	16,390	23,639	-30.5	16.35	38.8

the smaller drops. The statements of H. G. Houghton (2) that if a maximum number of small drops is required, nozzles of the smallest size practicable should be used and operated at the highest possible pressure are substantiated. The pressure differences required for *Type B* nozzles range from 10 to 47.5 lb per square inch gage for a water discharge rate from 835 to 2000 lb per hour.

Effect of Screen Packings in Spray Chamber: From the conclusion that most of the cooling takes place near the nozzles, it seemed reasonable to presume that an increase in the unit volume enthalpy conductance could be obtained by

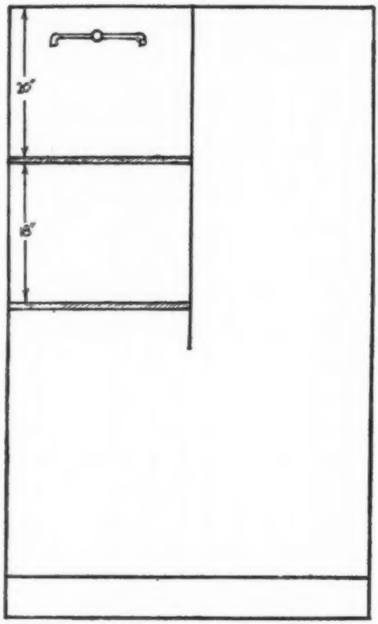


FIG. 13. ELEVATION SECTION OF SPRAY CHAMBER SHOWING THE POSITIONS OF THE SCREEN PACKINGS IN THE TOWER

placing wire mesh screen packing within the spray chamber perpendicular to the direction of the spray with the first screen far enough away from the nozzles to allow the fine drops to cool before striking the screen. After striking the screen, the lower portion of the tower would behave as a packed tower. The drops on leaving the nozzle cool rapidly at first and soon approach a limiting temperature. Due to the thermal resistance within the drops, a temperature gradient is established along the diameters with the center of drops being the hottest. When the drops strike the screen, mixing among the drops occurs and new but larger drops are formed on the underside of the screen and fall by gravity to the next screen where the drops are again mixed. This mixing of the water and forming of new drops presents a surface of higher temperature to the surrounding air, thus increasing the rate of enthalpy trans-

fer. Also, the time of contact between the water and air is lengthened. From a practical point of view, the introduction of the screens into the air stream will require more power to maintain the same air flow rate.

To determine the actual effect of screen packings within the spray chamber, a series of runs was made with two sections of packing and with screens of two different meshes (10). The inlet water temperature and gas rate were maintained approximately constant during the runs while the water rate was varied. The test data and results are presented in Table 4. Although the experimental points are scattered somewhat, the results are conclusive. The

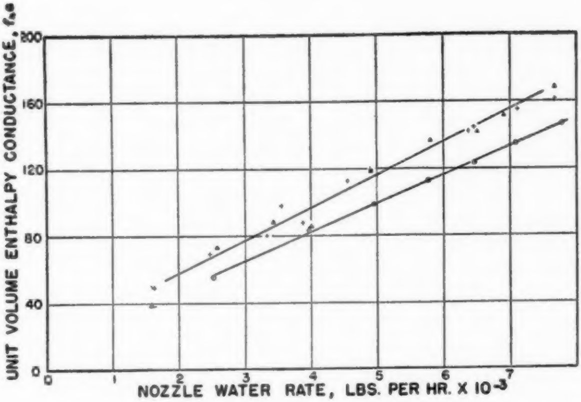


Fig. 14. Unit Volume Enthalpy Conductance *vs.* Nozzle Water Rate for Spray Chamber and Screen Packed Chamber

Inlet water temperature, 90 F. Dry air rate, 4600 lb per hour. Six nozzles, *Type A*. Legend: Triangle with point, 2 No. 4 mesh screens; cross, 2 No. 8 mesh screens; circle with point, without screens.

scattering is probably due to the fact that the data for these curves were taken during shorter runs and that various interfering factors within the laboratory made it difficult to maintain steady state conditions during the runs.

The conductance, f_{ha}, is plotted as a function of water rate in Fig. 14. The percentage increase in the enthalpy conductance ranges from 23 to 16 per cent for nozzle discharge rates from 3000 to 7000 lb per hour. Unfortunately, the power input to the fan was not measured during these series of tests so that the increase in cost due to the higher pressure drop in the tower could not be determined.

From the results of these tests, it is evident that the addition of the screen packings to the system improves the performance of the unit considerably but there is no apparent difference in performance by the use of the pair of finer mesh screens. The spray nozzles serve to form small drops to obtain the benefits of spray cooling and also distribute the water across the tower section.

The screen packings allow the droplets to intermingle and form surface sheets and larger drops as the water passes down the tower.

GENERAL CONCLUSIONS

A number of previous investigations have been made with respect to spray scrubbers or absorbers (8, 10, 12, 13) and correlations between analytical and experimental work have been partially established. Despite the similarity of the phenomena taking place within spray scrubbers and spray coolers (evaporative coolers), comparatively little work has been done in the latter field. Further investigations, experimental and analytical, will be required before a more complete correlation can be developed. The ultimate goal of these investigations is to establish a generalized basis for the correlation of the performance characteristics of the various types of transfer systems. It is visualized that these results, together with the reults previously obtained on other types of cooling towers (3, 15, 16), will be of future use for the classification of cooling towers on a generalized basis.

In conjunction with the experimental work on actual cooling towers, investigations have been made to more fully understand the microscopic phenomena of the various processes taking place within the towers (4, 9). As yet, it is difficult to attempt to obtain a good correlation between the two approaches due to the wide gap existing between the ideal and actual systems.

The following specific conclusions aid to summarize this work:

1. Within the operating range of the pressure spray nozzle *Type A* (Fig. 1 *a*), the discharge rate varies with 0.56 power of the pressure difference across the nozzles.

2. For the various conditions investigated, the unit volume enthalpy conductance varies approximately with the square root of the pressure at the nozzles or directly with the water rate.

3. For a constant nozzle discharge rate, constant gas rate, constant length of tower and inlet water temperature, only a slight increase in conductance can be obtained for a relatively high increase in nozzle pressure by decreasing the number of nozzles.

4. The greatest contribution to the over-all volume enthalpy conductance ($f_h aV$) is made in a region close to the nozzles, so that beyond a certain length of tower, the increase of this magnitude may be small.

5. For a constant gas rate, inlet water temperature, and pressure at the nozzles, the unit volume enthalpy conductance varies approximately with the −0.7 power of the tower length.

6. Although higher pressures are required, a higher conductance can be obtained by the use of a large number of small nozzles for a given water rate and tower volume. The limiting drop size depends on the gas velocity and must be such that no drift will occur (14).

7. A more effective tower results by the combination of the spray and screen packed types of cooling towers into a single unit.

8. The poor energy and material balances indicate substantial entrainment in the air stream (in general the energy and material balances are poorer at the higher water rates, *i.e.*, smaller drop sizes).

ACKNOWLEDGMENTS

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for their able assistance in the mechanical and experimental work. The information on spray nozzles made available by H. D. Bruce is also appreciated.

The cooperation of the Los Angeles and Golden Gate Chapters of the American Society of Heating and Ventilating Engineers, and the San Francisco Air Conditioning Society allows the continuation of this work. The tower and a set of spray nozzles were gifts of the W. R. Ames Co., San Francisco.

NOMENCLATURE

a = enthalpy transfer area per unit of spray volume, square feet per cubic foot of volume.
$f_d a$ = unit volume mass conductance, pounds per (hour) (cubic foot) (pounds per pound).
f_h = enthalpy transfer conductance, Btu per (hour) (square foot) (Btu per pound).
$f_h a$ = unit volume enthalpy conductance, Btu per (hour) (cubic foot) (Btu per pound).
$f_h a V$ = total volume unit conductance, Btu per (hour) (Btu per pound).
G = gas rate (dry air rate) through tower, pounds per hour.
h = enthalpy above the appropriate datum, Btu per pound.
Δh_{lm} = logarithmic mean enthalpy potential, Btu per pound of dry air

$$= \frac{(h_{T1} - h_1) - (h_{T2} - h_2)}{\log_e\left(\dfrac{h_{T1} - h_1}{h_{T2} - h_2}\right)}$$

h_T = enthalpy of saturated air at mixed mean liquid temperature T Btu per pound dry air.
h_{L2} = enthalpy of liquid (water) at outlet temperature.
L = liquid rate, pounds per hour.
L_1 = nozzle water rate (water on tower) pounds per hour.
ΔL = water evaporated into air stream measured as the difference of water in and out of the tower, pounds per hour.
H = absolute humidity, pounds per pound dry air.
ΔH = difference of absolute air humidity at exit and entrance pounds per pound dry air.
N = number of nozzles.
P = pressure at the nozzles, also the difference in pressure across the nozzle the tower pressure being substantially atmospheric, psig.
R = relative humidity of air, per cent.
S = tower cross sectional area, ft^2
t = air temperature, degree Fahrenheit.
T = water temperature, degree Fahrenheit.
ΔT = decrease in water temperature through tower, degree Fahrenheit.
V = spray volume (the volume of the tower), cubic feet.

Subscripts:

1 = conditions at water entrance section.
2 = conditions at water exit section.
lm = logarithmic mean.
DB = dry-bulb.
WB = wet-bulb.
da = dry air.
L = liquid (water).
$\log_e$ = logarithm to the Naperian base.

Properties of air-water vapor mixtures were obtained from the 1943 Edition of the Heating, Ventilating, Air Conditioning Guide.

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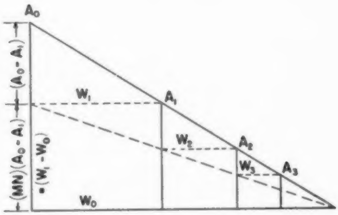
DISCUSSION

W. H. CARRIER, Syracuse, N .Y.: In 1908, A. E. Stacey and I ran a series of tests on a vertical spray air conditioning device which we used both for cooling the air with cold water and as in a cooling tower, for cooling warm water with air. In these tests we reached some rather interesting conclusions which, I believe, have never been published.

Briefly, the experimental apparatus consisted of a vertical tower about 4 ft square and perhaps 15 ft high. Conditioned air was introduced at the bottom of this tower and distributed upward through an eliminator at the top. It was so designed that from one to four spray banks could be installed at any height and at any space. The supplied air was first passed through a conditioning device, which in cool weather provided air at any desired constant temperature, humidity and quantity. Distributing vanes were used at the bottom of the tower to obtain an approximately uniform velocity over the cross-section of the tower. The sprays were arranged in each case to spray downwards against the air flow so that there was a counter flow of air and water, as in a cooling tower. No baffles were used inside the tower other than those at the entrance and exit.

Several facts of general importance were determined in this test which are applicable to this paper. *First,* when the air quantity, the water quantity and the entering wet-bulb temperature of the air were constant, the leaving wet-bulb temperature remained constant regardless of the variation in dry-bulb temperature. *Second,* with a single bank of sprays opposed to the direction of air flow, the optimum cooling

effect was produced at a height of about 6 ft above the point of air entrance, and at this spacing and at this elevation with a single row of sprays, the leaving wet-bulb temperature of the air and the leaving temperature of the water were practically identical. This held true for a spray concentration of from 2 to 5 gal of spray per square foot of cross-section and from 400 ft to about 600 fpm air velocity. It seems to hold for all sizes of nozzles used and for all nozzle water pressures of 20 lb or more. *Third,* after a second bank of sprays was placed 2½ or more feet above the first bank and supplied with an equal quantity of water at the same pressure, the effect was identical to that obtained by having two separate washers in series with the air flow, and so on for each additional bank, providing the spacing of the successive banks was at least 2½ ft. As many as four banks of nozzles were thus tested. With two or more banks of nozzles the average temperature of



$$\therefore\ A_0 - W_0 = (1+3M)(A_0 - A_1) = \frac{(A_3 - W_0)}{\left(\frac{3M}{1+3M}\right)^3}$$

WHERE N = 3 FOR EXAMPLE

FIG. A. DIAGRAM FOR DERIVING FORMULA FOR D_n

the leaving water was higher than the wet-bulb temperature of the leaving air. The relation of the entering and leaving water temperature and the entering and leaving air temperature could be very closely predicted by the following formula:

$$D_n = (A_n - W_o) = \left(\frac{NM}{1+NM}\right)^N (A_o - W_o) = \frac{(NM)^N}{(1+NM)^{N-1}} (A_o - A_n)$$

A_o = Initial air temperature
A_n = Final air temperature
W_o = Initial water temperature
W_n = Final water temperature
N = Number of banks of nozzles in vertical counterflow

$$M = \frac{\text{Average water temperature rise}}{\text{Total air temperature drop}} = \frac{\left[\frac{(W_1 - W_o) + (W_2 - W_o) + \ldots (W_n - W_o)}{N}\right]}{A_o - A_n}$$

D_n = Temperature difference between leaving air and entering water

This approximate formula was arrived at by the assumption of an average variation in heat content per degree of wet-bulb change cooling or heating the air between two definite limits. If the final air temperature was not exactly known, variation of the average heat content of the air per degree wet-bulb could be determined by cut and try. The formula can easily be derived by Fig. A, which is based on the assumption of uniform rate of air change in relation to water change. This, of course, is not strictly true, but the final results may be approximately the same,

within the limits of experimental error, as if it were true. This is a very convenient relationship to know in predicting performance of towers of this character.

It might be well to remark in this connection that the best experimental procedure for the solution of practical engineering problems is to first find results for ideal conditions and then determine the degree of deviation from these ideal conditions in practical applications. As, for example, in this case results were determined with uniform air distribution and then the deviation was determined from these results by unfavorable air distribution as often occurring in actual installations. This deviation can usually be represented as an efficiency factor and it is as important to know this efficiency factor as it is to know performance under ideal conditions. For example, with a single row of horizontal sprays opposed to the air flow, as in a horizontal air conditioner, the efficiency factor based on the ideal counter flow of the vertical tower lies between 80 per cent and 85 per cent. In a horizontal spray air conditioner with the nozzles discharging in the direction of air flow, the efficiency or application factor is considerably lower . . . in the neighborhood of 60 to 65 per cent for a single row.

The *humidifying* efficiency, however, with recirculated water and nozzles opposed to the direction of air flow may be made very much higher than this by the use of small nozzles with high pressure since in humidifying it is the fineness of the sprays rather than the volume of spray that is effective in producing saturation. Saturation efficiencies of 95 per cent or more can thus be produced with a single row of spray nozzles. The fine spray has little, if any, advantage beyond a certain point in producing actual heat transfer from colder water to warmer air or vice versa.

B. M. WOODS, Berkeley, Calif.: I agree with Dr. Carrier's belief that the basic experiments on the operation of any equipment should be conducted for reproducible conditions, that is to say, as nearly as possible ideal conditions. I am not certain about the nature of the flow, as to whether it is a well-distributed, directed flow in this tower. It was a counter-flow operation, of course, and I believe that baffles were in place. I should be greatly surprised if they were not. We have used them a good deal in the laboratory, but I cannot certify in this case.

You will note, in the computations that the mass balance and energy balance variations were indicated at every point, and that they went from virtually zero up to something like 10 per cent. The author discussed the difficulties of the practical problems involved. There were some of these with higher pressures and close spacing of nozzles. There was some carryover of moisture which escaped in the air stream, droplets and the like; and, of course, that upsets conditions. However, I think you will observe that values were moderately close to balance. I think it would be very helpful indeed if Dr. Carrier has some data on the subject that he would be willing to let Professor Boelter have, data from his files, because I know that the material would be appreciated.

1241

No. 1241

HEAT TRANSMISSION THROUGH INSULATION AS AFFECTED BY ORIENTATION OF WALL

By Frank B. Rowley * and C. E. Lund,** Minneapolis, Minn

The laws governing the thermal conductivity of insulating materials and of insulated walls have for most conditions been rather definitely established. It is usually possible to calculate with reasonable accuracy the overall thermal conductivity of a wall, providing the thermal properties for the various materials of construction are known. There are, however, some conditions under which the calculated results may be questioned. This may be due to some indefiniteness as to the properties of the materials or to differences in methods used in making the calculations.

Take, for example, a wood structure in which insulating material is placed between the framing members. For practical purposes in making the calculations it is often assumed that the insulating material covers the full area of the wall and the heat resistance through the studs is considered the same as that through the insulation. For relatively thin layers of insulation, this method would result in a calculated conductivity value for the wall which would be too high, whereas for relatively thick layers of insulation, it would result in a calculated overall conductivity which would be too low. For certain thicknesses of insulation in which the insulating value of the material between the studs would be equal to that of the studs, the calculated results should check with the test results. A more accurate method of calculation is to consider parallel heat flow through the framing members and through the insulation. In making calculations on this basis, the question arises as to how effective the full thickness of the framing members may be. One method is to assume a thickness equal to that of the insulation, and another is to use the full thickness of the framing members. Neither of these methods is absolutely accurate for all thicknesses of insulation in combination with a given thickness of studs or framing members.

Another question that may be raised in connection with porous or fill insulation is that of the possible effect of convection currents on the apparent thermal conductivity of the material and as to whether such possible convection currents might affect the heat flow differently for different orientations of the wall and for different directions of heat flow through the wall.

Density and fibre size of insulating material are also factors which affect their thermal conductivity, but for practical purposes, average values must be used in calculating the coefficients of insulated walls.

The purpose of the research reported in this paper was to determine some of these factors and to compare tests with calculated results for different appli-

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cations of insulating material in framed walls and for different orientations of the wall. The results here reported are confined to mineral wool insulation; however, other types of insulating materials were included in the tests, and the comparative results between calculated test values were found to be the same. In making the tests a standard hot-box apparatus was used and was so constructed that it could be rotated to place the wall either in a horizontal position with heat flowing upward or in a vertical position with heat flowing horizontally. Frame walls were used with both 2 x 4 in. and 2 x 6 in. framing members, and in some instances the framing members were omitted over the test

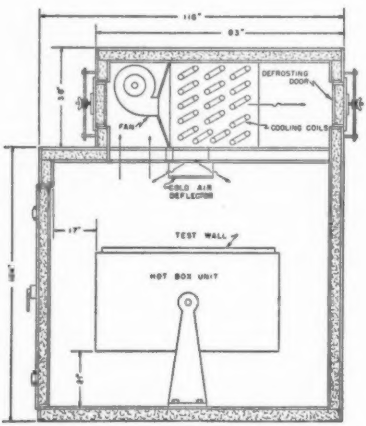

Fig. 1. Cross Sectional View of Cold Room Showing Test Box in Position for Conducting Heat Vertically Upward Through Wall

area to eliminate their effect upon the thermal value of the insulation. Mineral wool was used in granular, bat, and blanket form, and in thickness varying from 2 in. to 6 in. Various methods of application were used as described later.

Test Apparatus

The test apparatus consisted of a standard or guarded box construction. It was designed for a test wall 5½ ft square, mounted on trunnions so that the wall could be rotated to either a vertical or horizontal position, and placed in a cold room, the temperatures of which could be controlled to −10 F. Fig. 1 shows a cross-sectional view of the cold room with the test equipment installed. Fig. 2 gives the details of the test box with the test wall in place, and Fig. 3 is a photographic view showing the control panel and instruments located outside of the test box.

Referring to Fig. 1, which is a cross-sectional view through the cold room showing cooling coils and circulating fans above, the hot-box is shown in a horizontal position with test wall in place. The low temperature in the room is maintained by circulating the air through the cooling coils and controlling the temperature through the compressor.

Fig. 2 shows the plan for the test box with the wall removed, together with the two sectional views with test wall in position. The sectional views show the arrangements of the inner and outer test boxes, the heating element, the fans for circulating the air within both boxes, and the location of the various thermocouples used in the test apparatus, and throughout the test wall. Heat is furnished to the outer box by electric heating elements shown at the bottom of the sectional view, and the air is circulated by a fan as shown. Heat is supplied to the inner box by heating elements which are encased in a circulating duct, the warm air being circulated by a fan over the surface of the test wall.

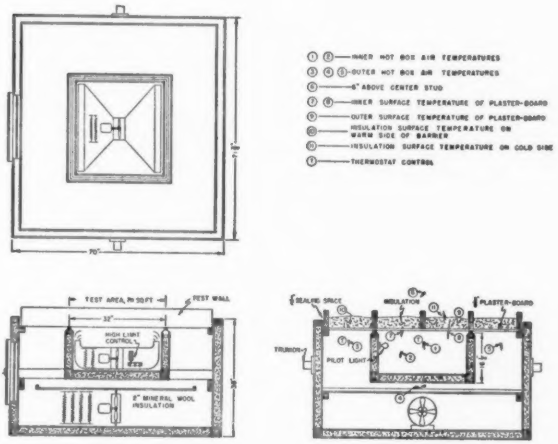

FIG. 2. DETAILS OF TEST BOX WITH TEST WALL IN PLACE

The temperatures of the air within the two boxes are controlled to 70 F by two constant temperature regulators which in turn control the electrical input to the heating elements. All of the current supplied to the inner heating box is metered and used in calculating the thermal coefficients of the walls. All of the temperatures are taken with copper constantan thermocouples and Leeds & Northrup potentiometer. The inner or test box proper is 32 in. square in order to coincide with the space between two 16 in. spaced framing members, and thus give average results over the test area including the studs or joists with the insulated section. All thermocouples and electrical leads are carried to the outside and connected to the control panels shown in Fig. 3.

TEST PROCEDURE

In making a test, a 5½ ft square wall was constructed and placed in position to seal the two open faces of the inner and outer boxes as shown in Fig. 2. The test box was then set in either a vertical or horizontal position as required for the test, and the temperatures within the inner and outer test boxes were brought to 70 F by the automatic temperature control and that in the cold

room was brought to −10 F by a separate automatic control. These temperature conditions were maintained until uniform temperature gradients were established throughout the test wall and box, after which records were kept of wall temperatures and of the heat supplied to the inner test box for a sufficient length of time to assure uniform and accurate test conditions. The uniform temperature period usually averaged eight hours, although it was not definitely set. From the constant temperature and electrical meter reading, the thermal conductivity coefficients were calculated.

DESCRIPTION OF WALLS TESTED

Insofar as the framing construction is concerned, all walls tested may be divided into three groups. Walls of Group 1 were constructed with 2 x 6 in. framing members spaced 16 in. on center with a 2 x 6 in. member at each end of the wall. The inner or warm surface was covered with ⅜ in. gypsum plaster board. This construction was tested without insulation between the framing members and with various thicknesses of insulation applied as described for the various tests within the group.

Walls for Group 2 were constructed without the central framing members, that is with 2 x 6 in. joists only at the extreme edges at the top and bottom of wall, and with ⅜ in. gypsum plaster board on the warm side. This wall was tested with various thicknesses of granular mineral wool hand-applied over the gypsum board in order to get the effect of various thicknesses in insulation without the effect of the framing members.

The wall of Group 3 was of the same construction as those of Group 1, with the exception that 2 x 4 in. framing members were used in place of the 2 x 6 in. framing members of the first group. The framing members were cut to exactly 4 in. in thickness.

METHOD OF APPLYING INSULATION

When granular mineral wool was applied by hand, it was poured into the space between the framing members and levelled off as closely as possible to the required thickness with a strike. The exact thickness was measured by use of a special gage, the various points of which could be adjusted for the exact thickness of insulation. The average thickness for the test was taken as the gage thickness for approximately 60 points over the test area.

In applying the machine-blown mineral wool to the open construction—that is, without flooring, and in a horizontal position—the open end of the blowing hose was held horizontal and about 24 in. above the upper surface of the joists. The space between was filled as closely as possible to the desired thickness, after which it was levelled off with a strike, and the exact thickness measured by the special gage. For the horizontal wall with 4 in. joists in Group 3, the mineral wool was blown level with the joists. For the wall of test 36 with 6 in. joists, the mineral wool was blown to a thickness of 4 in. after which the floor boards were applied with screws. For the wall of test 37 with 6 in. joists, the floor was in place and the space between the joists was blown full with granulated mineral wool.

For the mineral wool bats in tests 27 and 39 with barriers designated as lapped on the warm side of the joists, the paper flaps along the edges of the

bats were lapped over and nailed to the lower faces of the joists underneath the gypsum plaster board. For tests 21 and 22 designated as "barrier laid on warm side," the bat was applied from the cold side of the wall without removing the plaster board, and the flaps were turned upward and pressed in contact with the vertical sides of the joists.

When the walls were tested in a vertical position with loose insulation between the framing members, it was necessary to apply a wire screen over the top of the insulation in order to hold it in place when the wall was put in a vertical position. For tests 5, 6, 10, 11, 16, 17, and 18, a ¼ in. mesh wire

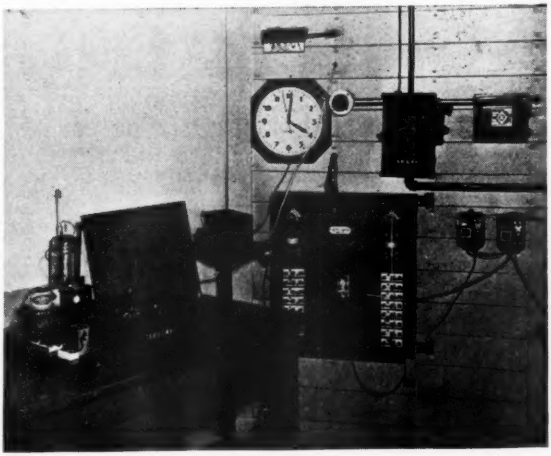

FIG. 3. CONTROL PANEL AND TEST INSTRUMENTS

covering was fastened between the joists to support the insulation. In order to get true comparative results, the walls were tested with the wire in place both in the horizontal and vertical position.

DISCUSSION OF TEST RESULTS

A summary of test results is given in Table 1. The temperatures as taken throughout the wall and used for making the calculations are given in Table 2. Referring to Table 1, the walls tested are divided into three groups. The numbers of tests made on each wall of the group are indicated in the third column. When more than one test was made on a given wall, the averages of all of the tests were used for calculating the various coefficients. The fourth column of the table gives the orientation of the wall as tested, and when tested in a horizontal position, the heat was always flowing upward. The thickness and density of the insulation as given in columns 5 and 6, were taken of the insulation as applied in the wall. Thicknesses under the description of test section are the nominal thicknesses and not the actual thicknesses as tested. The thermal conductivity coefficient, *k*, under the column headed *Test* were calculated from

TABLE 1—SUMMARY OF TEST RESULTS

TEST NO.	DESCRIPTION OF TEST SECTION	NO. OF TESTS	WALL POSITION	INSULATION		CONDUCTIVITY k[a]		U VALUE	
				Thickness Inches	Density Lb/Cu Ft	Test[b]	Corrected[c]	Test	Corrected[d]
GROUP 1: ⅜ IN. GYPSUM PLASTER BOARD, 2 × 6 IN. JOISTS									
1, 26, 33	No insulation	3	Horizontal	...	...	...	...	0.817	0.938
19	2 in. Hand applied granular mineral wool	1	Horizontal	2.12	6.05	0.340	0.288	0.139	0.137
2, 3, 4, 12	3 in. Hand Applied granular mineral wool	4	Horizontal	3.15	6.60	0.341	0.289	0.102	0.101
5	3 in. Hand Applied granular mineral wool with wire covering	1	Horizontal	3.00	6.62	0.343	0.291	0.101	0.102
6	3 in. Hand Applied granular mineral wool with wire covering	1	Vertical	3.00	6.62	0.343	0.291	0.098	0.100
35	3 in. Machine blown granular mineral wool	1	Horizontal	3.21	5.90	0.308	0.252	0.096	0.087
7, 8	4 in. Hand Applied granular mineral wool	2	Horizontal	4.05	6.14	0.373	0.326	0.084	0.084
10	4 in. Hand Applied granular mineral wool with wire covering	1	Horizontal	4.05	6.14	0.359	0.309	0.082	0.081
11	4 in. Hand Applied granular mineral wool with wire covering	1	Vertical	4.05	6.14	0.359	0.309	0.080	0.081
9	4 in. Hand Applied granular mineral wool with paper covering	1	Horizontal	4.05	6.14	0.369	0.320	0.080	0.083
14, 15	4 in. Machine blown granular mineral wool	2	Horizontal	4.04	5.57	0.358	0.308	0.082	0.081
16, 18	4 in. Machine blown granular mineral wool with wire covering	2	Horizontal	4.07	5.70	0.341	0.289	0.076	0.077
17	4 in. Machine blown granular mineral wool with wire covering	1	Vertical	4.09	5.84	0.343	0.291	0.077	0.077
36	4 in. Machine blown granular mineral wool with 1 in. flooring	1	Horizontal	3.87	5.82	0.288	...	0.069	0.069
34	4 in. Machine blown loose mineral wool	1	Horizontal	4.14	6.39	0.377	0.329	0.083	0.083

37	6 in. Machine blown granular mineral wool with 1 in. flooring	1	Horizontal	5.125	7.27	0.323	...	0.059	0.059
39	3 in. Mineral wool bats, barrier lapped on warm side	1	Horizontal	3.478	3.67	0.346	0.295	0.092	0.090
21	3 in. Mineral wool bats, barrier laid on warm side	1	Horizontal	2.646	2.24	0.316	0.261	0.106	0.106
22	3 in. Mineral wool bats, barrier laid on warm side	1	Vertical	2.646	2.24	0.303	0.247	0.102	0.102
27	4 in. Mineral wool bats, barrier lapped on warm side	1	Horizontal	3.89	2.99	0.359	0.309	0.085	0.084
28	4 in. Mineral wool bats, barrier lapped on warm side	1	Vertical	3.89	2.99	0.378	0.330	0.088	0.088
20	4 in. Mineral wool bats, no barrier	1	Horizontal	3.99	1.77	0.354	0.304	0.082	0.081
	GROUP 2: 3/8 IN. GYPSUM BOARD, JOISTS OMITTED IN TEST SECTION								
30	2 in. Hand Applied granular mineral wool	1	Horizontal	1.906	6.36	0.303	...	0.136	0.139
46	3 in. Hand Applied granular mineral wool	1	Horizontal	2.635	6.80	0.311	...	0.107	0.107
32	3 in. Hand Applied granular mineral wool	1	Horizontal	2.940	6.05	0.327	...	0.101	0.101
29	4 in. Hand Applied granular mineral wool	1	Horizontal	3.875	6.67	0.307	...	0.075	0.074
48	6 in. Hand Applied granular mineral wool	1	Horizontal	5.125	7.13	0.325	...	0.058	0.060
	GROUP 3: 3/8 IN. GYPSUM BOARD, 2 IN. × 4 IN. JOISTS 4 IN. DEEP								
38	4 in. Machine blown granular mineral wool	1	Horizontal	3.625	5.76	0.356	0.303	0.089	0.089

[a] Conductivity k is based on differences of overall test coefficients of insulated and non-insulated walls.
[b] Test coefficient includes studs and is based on thickness of insulation as applied.
[c] Corrected on basis of parallel heat flow through insulation and joists and on actual thickness of insulation as applied.
[d] Corrected to still air on inside surface and 15-mile wind velocity on outside surface of wall.

TABLE 2—TEST RESULTS

Test No.	Date	Air Temperatures, deg F			Surface Temperature, deg F					Overall Coef. U as Tested
					Plaster Board		Insulation		Flooring	
		Inner Box	Outer Box	Outside	Warm Side	Cold Side	Warm Side	Cold Side	Cold Side	
1	2/11	70.1	71.6	−9.1	42.6	23.2	...	...	...	0.803
26	3/25	70.1	70.8	−9.0	43.6	24.8	...	...	...	0.804
33	4/27	70.3	70.9	−9.2	43.5	24.7	...	...	...	0.843
19	3/10	70.0	70.3	−10.5	66.4	63.7	...	−6.7	...	0.139
2	2/12	70.2	70.3	−10.7	67.4	65.6	...	−8.4	...	0.097
3	2/13	70.1	70.2	−10.3	67.3	65.5	...	−8.2	...	0.105
4	2/15	70.1	70.4	−9.4	67.4	65.2	...	−6.9	...	0.104
12	2/26	70.1	70.4	−10.5	67.6	65.7	...	−7.9	...	0.103
5	2/17	70.0	70.3	−10.6	67.5	65.8	...	−6.1	...	0.101
6	2/18	70.2	70.5	−6.2	67.8	66.1	...	−1.2	...	0.098
35	4/29	70.1	70.3	−10.0	67.8	65.7	...	−6.1	...	0.096
7	2/20	70.1	70.3	+0.1	68.5	67.3	...	+3.3	...	0.081
8	2/22	70.0	70.5	−10.5	68.1	66.7	...	−7.9	...	0.088
10	2/24	70.1	70.7	−10.6	68.1	66.7	...	−8.1	...	0.082
11	2/25	70.1	71.2	−11.4	68.2	67.0	...	−8.1	...	0.080
9	2/23	70.0	70.6	−10.5	68.1	66.8	...	−5.3	...	0.081
14	3/2	70.0	70.3	−10.4	68.0	66.6	...	−8.6	...	0.082
15	3/3	70.0	70.3	−10.3	68.1	66.6	...	−8.4	...	0.082
16	3/5	70.1	70.3	−10.3	68.1	66.7	...	−6.1	...	0.073
17	3/6	70.1	70.6	−9.8	68.2	67.0	...	−6.5	...	0.077
18	3/8	70.1	70.4	−10.3	68.2	66.8	...	−8.6	...	0.080
36	4/30	70.1	70.3	−10.0	68.5	67.0	...	+3.6	−7.3	0.069
34	4/28	70.1	70.3	−10.3	68.0	66.2	...	−7.3	−7.8	0.083
37	5/2	70.3	70.3	−10.2	69.0	68.0	...	−0.7	...	0.059
39	5/4	70.2	70.3	−10.5	68.6	66.1	...	−8.1	...	0.092
21	3/16	70.2	70.3	−10.2	67.4	65.5	63.7	−6.8	...	0.106
22	3/18	70.0	70.5	−10.2	67.6	66.2	63.9	−6.1	...	0.102
27	3/22	70.0	70.2	−10.3	68.0	66.5	63.8	−7.8	...	0.085
28	3/30	70.0	70.2	−10.8	67.9	66.6	63.8	−8.0	...	0.088
20	3/13	70.1	70.3	−10.3	68.1	66.8	...	−7.8	...	0.082
30	4/1	69.9	70.0	−10.2	66.0	62.6	...	−4.3	...	0.136
32	4/26	70.2	70.4	−10.2	67.1	64.5	...	−7.3	...	0.101
46	5/12	70.0	70.2	−10.3	67.5	64.0	...	−5.4	...	0.107
29	3/30	70.0	70.1	−10.2	67.9	66.1	...	−8.1	...	0.075
48	5/13	70.0	70.1	−10.8	68.5	66.7	...	−8.0	...	0.060
38	5/3	70.5	70.3	−10.2	68.5	66.7	...	−6.6	...	0.089

the differences in heat resistance for the insulated and uninsulated walls, from the thickness of insulation applied between the framing members and on the assumption that the insulation covered the full area of the wall. Coefficients under the column headed "Corrected" have been corrected to take into consideration the areas as covered by the insulation only. The overall coefficients *U* under the ninth and tenth columns are first those calculated from test values from air to air, and second the corrected values based on coefficients for still air on the warm side of the wall and for air at a 15-mile wind velocity on the cold side of the wall.

In determining the test surface coefficients, it was necessary to measure surface temperatures. Although great care was used in measuring the surface temperatures of the exposed surfaces of the insulation, there was still some uncertainty as to what the exact values should be, and since they were used in the calculation of surface coefficients, there may be some slight errors in the corrected value for *U*. However, these are not great.

EFFECT OF WALL POSITION ON THERMAL CONDUCTIVITY

The comparative results for walls tested with heat flowing upward in a horizontal position and with heat flowing horizontally with the wall in a vertical position are shown by the results of three different walls which were tested both in horizontal and vertical positions. For the first wall, test 5 with the wall in a horizontal position, $U = 0.102$, and for test 6 with the same wall in a vertical position, $U = 0.100$. For the second wall, test 10 with the wall in a horizontal position, $U = 0.081$. For the same wall in a vertical position as in test 11, the *U* value is the same. For the third wall in a horizontal position, the average of tests 16 and 18, $U = 0.077$. The same result was obtained for test 17 with the wall in a vertical position. Each of these walls had a wire netting applied over the insulation for both tests. These tests indicate that there is no appreciable difference in thermal conductivity for the type of walls tested when placed either in a vertical or in a horizontal position.

EFFECT OF THICKNESS OF INSULATION

The effect of the insulation thickness on the thermal conductivity value, *k*, with heat flowing upward through the insulation, is shown by the results of the tests in Group 2. The results of these tests and the method of obtaining the thermal conductivity value, *k*, for the insulation have been summarized in Table 3. In making the tests, a wall without framing members over the central section was used. That is, the loose or granulated mineral wool was hand applied to the upper surface of the plaster board over the full test area without joists. It was applied in various thicknesses which were measured by the special test gage provided for the purpose and given in Table 3. All walls were of the same construction with the exception of thickness of insulation applied, and in each case the same type of granular mineral wool was applied. The overall coefficient for the gypsum plaster board only, which represents the wall without insulation, was calculated from test values which were obtained on other test apparatus. However, the calculated value corresponds with the test value for the uninsulated wall with framing members after a correction is made for these members. The calculated value was used in order to obtain

from test results the actual resistance of the insulation applied to each of the other walls.

Referring to Table 3, the overall coefficients U are the test values for all but the first wall without insulation. The overall resistances are the reciprocals of these coefficients, and the net resistance due to the insulation represents the difference between the heat resistance of the wall as tested and the resistance of the wall without insulation. The thermal conductivity values k for the insulation in the last column are based on the net resistance and the actual thickness of the insulations as applied. These values are reasonably consistent, considering the fact that it is impossible to make exact determinations of the thickness of the insulation and also that there may be a variation in the size of pellets which make up the insulation. There is no trend toward higher conductivity values per inch of thickness as the thicknesses increase.

TABLE 3—EFFECT OF THICKNESS OF INSULATION ON THE THERMAL CONDUCTIVITY k WITH HEAT FLOWING UPWARD

Test No.	Insulation	Actual Thickness of Insulation Inches	Overall Coefficient U	Overall Resistance $1/U$	Net Resistance of Added Insulation	Conductivity k for Added Insulation
1, 26, 33	None	....	1.10	0.909	....	....
30	2-in. Hand Applied Granular Mineral Wool	1.906	0.139	7.194	6.285	0.303
46	3-in. Hand Applied Granular Mineral Wool	2.625	0.107	9.345	8.436	0.311
32	3-in. Hand Applied Granular Mineral Wool	2.940	0.101	9.901	8.992	0.327
29	4-in. Hand Applied Granular Mineral Wool	3.875	0.074	13.514	12.605	0.307
48	6-in. Hand Applied Granular Mineral Wool	5.125	0.060	16.667	15.758	0.325

EFFECT OF STUDS ON THE APPARENT VALUE OF INSULATION

From the test results for walls with and without studs, the differences in the overall value for the walls become greater as the thickness of insulation is increased. If curves are plotted, it will be found that the overall coefficients are substantially the same for 2 in. of insulation either with or without studs. For 4 in. of insulation the average U values from the curves are 0.072 for the wall without studs and 0.084 for the wall with studs. This indicates the differences or errors that might be made by assuming the insulating material to cover the full area of the wall and not corrected for the framing members.

EFFECT OF PAPER COVER OVER SURFACES OF INSULATION

A comparison of the results for test 9 and the averages for tests 7 and 8 shows the effect of laying a heavy kraft paper over the top and in contact with

the insulating material. With the paper in contact with 4 in. of insulation between the studs, the overall coefficient is 0.083, whereas without the paper it was 0.084. The paper was laid over the insulation in contact with the surface, and fitted closely between the joists.

COMPARISON OF VALUES FOR HAND-APPLIED AND MACHINE-BLOWN INSULATION

For test 7 and test 8, 4.05 in. of granular insulation placed between the studs gave an overall coefficient of 0.084. For tests 14 and 15, 4.04 in. of granular insulation machine-blown between the studs gave an overall coefficient of 0.081. For test 34, 4.14 in. of loose wool insulation blown between the studs gave an overall coefficient of 0.083. In all of these tests, the densities were from 6.3 to 6.6, and the results were substantially the same. For test 35, however, 3.21 in. of granular rock wool blown between the joists gave an overall coefficient of 0.087. In this case, the apparent insulating value of the material is better than it was in either of the other cases. However, in this case the density is 5.90, which may explain the difference. In general, it appears that for the same type of material, the results for the hand-applied material will be substantially the same as those for the machine-blown material.

All the walls except those for tests 36 and 37 were tested without attic flooring. For test 36, 3.87 in. of blown granular mineral wool was covered with 1 in. flooring. This wall, tested in a horizontal position, gave an overall coefficient of 0.069 which corresponds almost exactly with the calculated value when correcting for the studs. Comparing this value with figures given in the Heating Ventilating Air Conditioning Guide 1943, the closest construction for a ceiling is one consisting of metal lath and plaster, 3⅝ in. rock wool fill insulation, and yellow pine flooring on joists. The coefficient *U* given for this construction is 0.068. Thus, there is no substantial difference between the test value and that used in The Guide.

DISCUSSION

R. S. DILL, Washington, D. C. (WRITTEN): I do not attach as much importance to the orientation of the wall as the authors appear to do for the reason that if convection occurs in the wall when the wall is in a horizontal position, convection will also occur in the wall when it is in a vertical position, although admittedly the amount of the convection may not be the same for the two cases. Since insulation is not ordinarily left uncovered when installed in vertical walls, the significance of the observations on materials in this position decreases further in my opinion.

I am in agreement with Professor Rowley that the preponderance of the data and information which has come to hand so far indicates that the effect of convection currents in horizontal layers or blankets of insulation is practically insignificant and furthermore that in computing the heat transfer through walls, the effect of timbers in the form of joists or other house elements which extend through the insulating layer should be taken into account.

In their paper the authors use this sentence: "One method is to assume a thickness equal to that of the insulation, and another is to use the full thickness of the framing members." By thickness is meant that dimension, of a framing member, parallel to the heat flow.

In my opinion, use of the full thickness of the framing members is not justified if such members project very much above the insulating material. On the other hand, if the members do project above the insulating material, their insulating effect may be slightly underestimated if their thickness is assumed to be exactly equal to that of the insulating material. In the present state of knowledge, I would favor an assumption of a thickness ¼ in. greater than that of the insulating material for purposes of computation when the members project more than ¼ in. above the insulating material.

Neither of the above methods of computation or any other approximation is absolutely accurate. There are lateral gradients to the heat flow through a combination of wooden members and insulating material such as that under consideration here, which prevent the computations from being in the class of absolute accuracy. Such computations, however, are considered to be sufficiently accurate for their purpose.

The data obtained by Professor Rowley and his co-author, as well as the results of some less formal tests made here, indicate strongly that the convection effect does not require consideration for practical purposes. However, as long as the opposition view has the backing of Prof. G. B. Wilkes, Massachusetts Institute of Technology, perhaps some work on the subject should continue until the matter is brought to a conclusion satisfactory to all concerned; but in the meantime my recommendation is that convection be ignored so far as estimates of heat losses from buildings are concerned.

T. T. TUCKER, Atlanta, Ga. (WRITTEN): The stones which compose the foundation of our Society are the technical data which we publish for the benefit of mankind. The mortar which cements these stones together is the fact that these technical data work. This paper is another proof of the practicality of our Society. It is a *refresher* course, so to speak, which we have passed.

But this paper goes further than that, it shows conclusively where many laymen have erred in trying to add things to insulation in the hopes of improving its effectiveness. For example, we see now what every one should have known all the time. Insulation properties do not change with position. Also, there is no difference in the practical results obtained from factory fabricated bats, mechanically applied materials, and hand packed granules. Contrary to the opinion of many architects and federal specification writers the paper backing on mineral wool has no bearing on its efficiency; regardless of how the paper is installed. The authors have by this splendid paper shown that the only thing which will improve good insulation is more insulation.

The authors have shown too that there is really no need for us to bog down in our own technical data. They have recognized the difficulty of determining the exact thicknesses of mineral wool. They have recognized too that things about which the ultimate user seldom bothers himself such as fiber diameter, pellet size and installed density do affect thermal conductivity. I quote "for practical purposes, average values must be used in calculating coefficients."

Perhaps the authors should answer these questions: Was the machine blown mineral wool applied in accordance with usual practice in the field and by some one experienced in that art? If so, how do they account for the density of this wool being so much greater than the factory fabricated bats which they tested?

L. R. VIANEY,[1] Cambridge, Mass. (WRITTEN): I am pleased to be able to be here to comment on the work presented by Professor Rowley.

There are several points which I would like to bring up for discussion and clarification.

Our experience at M.I.T. indicates that heat transfer measuring equipment with an on and off power input does not give as satisfactory equilibrium conditions as can be obtained with constant input type of apparatus. If I am not mistaken, the guarded

[1] **Instructor in Mechanical Engineering, Massachusetts Institute of Technology.**

plate *standards* call for a constant input to the central section and a differential thermocouple control of the guard ring. I believe that this should also be applied to the guarded box test thus eliminating the relatively large temperature differentials required to operate thermostatic controls.

Another point in the authors' equipment is the apparent lack of separating partitions between the insulation comprising the test area and that in the guard area. It has been our experience and also that of Professor Queer of Pennsylvania State College that such partitions prevent heat transfer between the guard and test areas by convection through the insulating medium or enclosed air spaces. These separators should in my opinion be incorporated into a test of the sort reported by Professor Rowley.

Furthermore there is some doubt in my mind as to whether or not sufficient time has been allowed after a change in test conditions for the equipment and insulating material to reach thermal equilibrium.

Since the air temperature in Professor Rowley's equipment was controlled by thermostats, I presume that the heaters and cooling units had sufficient capacity to bring the surrounding air to the required temperature rapidly. Since surface temperatures will quickly follow a change in air temperature, they also will be rapidly brought up to and maintained at the temperatures fixed and controlled by the thermostats. Therefore, apparently constant temperature gradients will be attained in a very short time while thermal equilibrium in the insulating material will not be reached for a considerable time after the surface temperatures are constant. Thus if changes in test conditions are made and readings are taken too soon thereafter, true steady state conditions will not exist and the resulting test values will be erroneous.

I would like to quote some recent work performed at M.I.T. to substantiate my previous statements. The following data were obtained with apparatus similar to Professor Rowley's except that our input to the test area was constant and that to the guard area was controlled by a multiple differential thermocouple and photoelectric relay arrangement. The actual operating temperatures were different.

Data from Tests at M.I.T.

Date	U[a]	Air Temp deg F	
		Inside	Outside
5/18	Installed	3 in. mineral wool	5 lb per ft³—no joists
5/19	0.126	. . .	. .
5/20	0.120	139	75
5/21	0.120	139	75
5/21 1 P.M.	Laid 1 layer paper over entire surface		
5/22	0.117	141	75
5/23	. . . .	. . .	. .
5/24	0.113	144	77
5/25	0.110	145	76
5/26	0.110	145	75

[a] U = Btu, hr^{-1}, ft^{-2}, deg F^{-1} Average of hourly calculations.

These results show in my opinion that even for the minor change which was made in the above test a very considerable time is required to reach true equilibrium conditions. In comparison I would like to point out a set of results taken from Professor Rowley's paper which may be found on following page.

TEST NO.	DATE	TYPE OF TEST
32	4/26	3 in. hand applied wool—no joists
33	4/27	No insulation
34	4/28	4 in. machine applied wool
35	4/29	3 in. machine applied wool
36	4/30	4 in. machine applied wool with 1 in. floor
..	5/1	No test recorded
37	5/2	6 in. machine applied wool with 1 in. floor
38	5/3	4 in. machine applied wool with 2 × 4 joists
39	5/4	3 in. mineral wool bats

It is my opinion that if tests are conducted as rapidly as indicated, we cannot be sure that true thermal equilibrium exists, and, therefore, if we are not sure of steady state conditions, we cannot be sure of the resulting coefficients.

G. L. TUVE, Cleveland, Ohio: There may be a considerable difference between laboratory results and the heat loss of a wall insulated in practice. One difference is in the scale size of the laboratory test panel and a full size wall. The insulation in the full size wall may not be applied as well as in the test panel.

Since many of the tests were run in February and March the materials were probably quite dry. Is there a possibility that moisture conditions obtained in an installation might affect the heat transmission to such an extent that some precaution should be given the man who applies the insulation?

R. K. THULMAN, Washington, D. C.: It is important to determine the correct heat transmission data especially for ceilings in order to prevent confusion and disappointment in the minds of the public which would purchase insulation. Since within six months two papers by eminent investigators have reported widely varying results and since these results varied from figures (taken from A.S.H.V.E. Guide) shown in FHA publications, it was important that the Society establish the correct figures which should be used.

AUTHORS' CLOSURE: Standard applicators with standard blowing equipment had been used for machine blown insulation and the author observed the work.

The density of average mineral wool as now manufactured is less than THE GUIDE value and there is a range of densities available. The conductivity values are substantially the same as given in THE GUIDE. Density is not the determining factor in conductivity which depends on fiber size and the pack of the fibers. Heavy fibers might weigh a lot and yet be very porous. Light material may test better than a heavier one or vice-versa. Continuous and on-and-off controls were found to give the same results when the controls operated properly.

In answer to Mr. Vianey's comment that the tests might not have been conducted under equilibrium conditions, all tests were conducted at equilibrium conditions held for 6 to 8 hours. Many of the tests were repeated with same results as reported. When the same walls were tested for more than one day no shut down occurred between tests. The effect of separators had been found by test to be negligible. The accuracy of the test methods employed had been proved by the ability to test-check day after day.

The effect of moisture would not be important in mineral wool because it is not hygroscopic. Even in hygroscopic material such as wood fiber material moisture would not be expected materially to affect test results. It would have no effect in comparative tests of two wall positions.

Test results were found to agree almost exactly with THE GUIDE figures for overall values for similar walls. THE GUIDE values therefore appear to be accurate, practical and perfectly logical for use.

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GRAPHICAL METHOD OF CALCULATING HEAT LOSSES

By PAUL D. CLOSE,* CHICAGO, ILL.

THE PURPOSE of this chapter is to present a graphical, time-saving method of calculating heat losses. Although the chart (Fig. 1) is applicable mainly to simple structures such as one, one and one-half, and two-story residences, the same principles can be adapted to other types of buildings.

This graphical method was developed on the basis of the fundamental equations for building heat losses. It is not intended to supplant the A.S.H.V.E. Guide procedure, but it may be useful for checking the Guide result in many cases, or for obtaining a reasonably accurate answer in a minimum of time. Since the calculation of building heat losses is not and cannot at best be an exact science, an approximate answer may suffice in many cases. The result obtained by this chart will generally check within ±3 per cent of the Guide result when the same coefficients and surface areas are used in both cases.

BASIS OF DERIVATION

The heat losses of a building are classified as (1) the transmission losses through the exposed surfaces such as the walls, ceiling, floor, glass, and door areas, and (2) the infiltration losses due to air leakage through cracks, crevices, and other openings. The transmission losses are a direct function of the areas of the exposed surfaces and their respective heat loss coefficients and the temperature difference. The infiltration losses are a function of the volume of outside air entering the building which must be translated into the heat equivalent required to warm this air from the outside temperature to the inside room temperature.

Symbols

The symbols used in the development of the formulae from which the curves of Fig. 1 were plotted are as follows:

U_w = coefficient of transmission of exterior walls (Btu per hr per sq ft per deg temperature difference);
U_g = coefficient of transmission of glass surfaces (Btu per hr per sq ft per deg temperature difference);
U_c = coefficient of transmission of top floor ceiling (plus roof) (Btu per hr per sq ft per deg temperature difference);
U_f = coefficient of transmission of floor (Btu per hr per sq ft per deg temperature difference);
A_1 = net wall area, sq ft (inside dimensions);
A_2 = net glass and door area, sq ft (inside dimensions);
A_w = gross wall area = $A_1 + A_2$ (inside dimensions);
A_c = ceiling area, sq ft (inside dimensions);
A_f = floor area, sq ft (inside dimensions);
H_d = heat loss of building per 1 deg temperature difference, Btu per hr.

* Technical Secretary, Insulation Board Institute. MEMBER of A.S.H.V.E.

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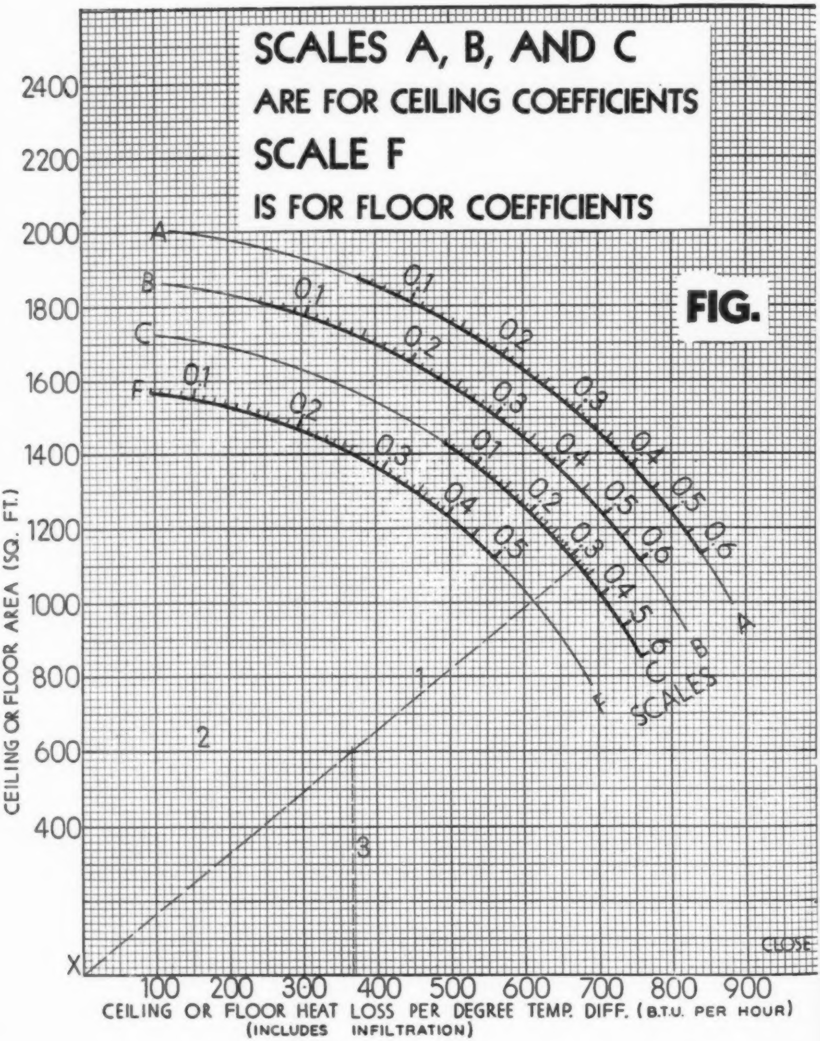
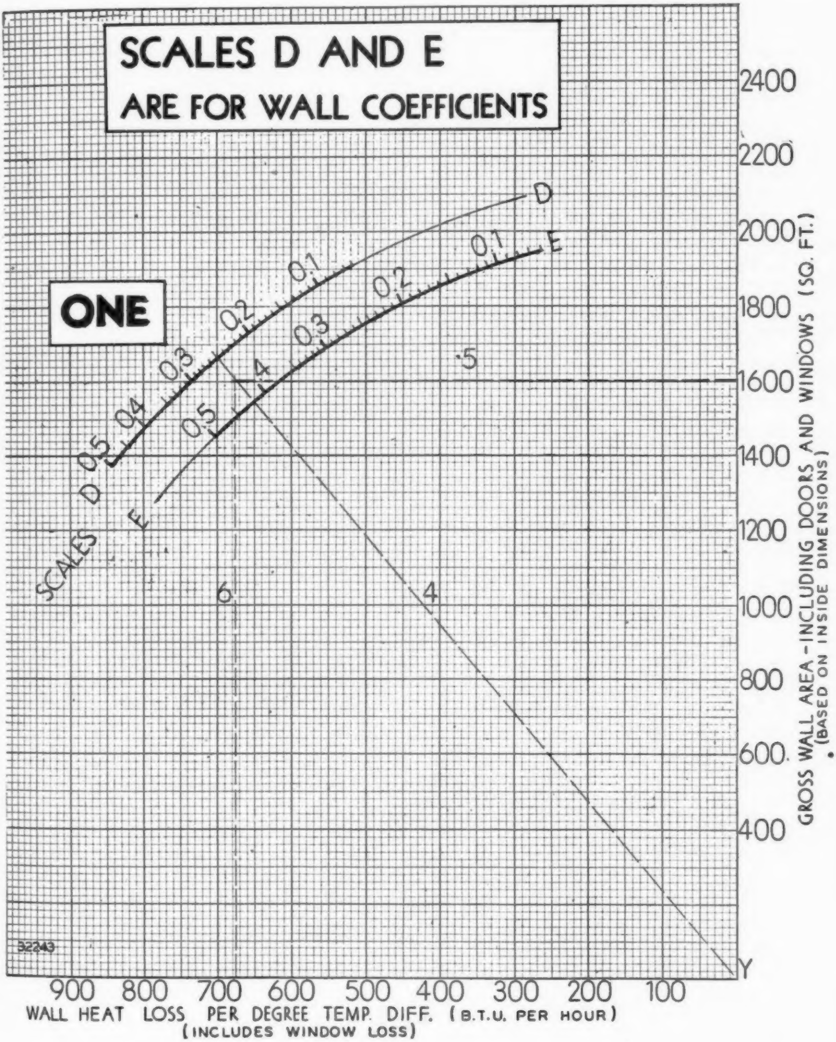

SCALES A, B, AND C
ARE FOR CEILING COEFFICIENTS
SCALE F
IS FOR FLOOR COEFFICIENTS
FIG.
CEILING OR FLOOR AREA (SQ. FT.)
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
X
100
200
300
400
500
600
700
800
900
CEILING OR FLOOR HEAT LOSS PER DEGREE TEMP. DIFF. (B.T.U. PER HOUR)
(INCLUDES INFILTRATION)
A
B
C
F
SCALES
1
2
3
CLOSE


SCALES D AND E
ARE FOR WALL COEFFICIENTS
ONE
SCALES D E
D
E
0.5 0.4 0.3 0.2 0.1
0.5 .4 0.3 0.2 0.1
5
6
4
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
Y
900 800 700 600 500 400 300 200 100
WALL HEAT LOSS PER DEGREE TEMP. DIFF. (B.T.U. PER HOUR)
(INCLUDES WINDOW LOSS)
GROSS WALL AREA - INCLUDING DOORS AND WINDOWS (SQ. FT.)
(BASED ON INSIDE DIMENSIONS)
32243

Transmission Losses

The wall and glass transmission losses *per degree temperature difference* are expressed by the following equation:

$$\textit{Wall and glass transmission losses} = A_1U_w + A_2U_g \quad . \ . \ . \ . \ . \ . \ . \quad (1)$$

It will be noted that the area factor (A_2) includes both the glass and door areas, whereas the coefficient factor (U_g) is for the glass heat loss coefficient. Actually the door coefficient will be lower in the case of solid wood doors than the glass coefficient, but since the door area is relatively small, the glass coefficient may be used for both the glass and door surfaces.

The glass and door areas of most residences range between 15 and 25 per cent of the gross exterior wall area (based on inside dimensions) or an average of 20 per cent. In order to simplify the procedure, this average glass and door area is assumed. A few trial calculations will show that there will be a comparatively small variation in the final result due to any small deviation from this average figure, provided the correct gross wall, glass and door area (A_w) is used. Substituting this ratio of glass and door area in equation 1, the wall and glass transmission loss becomes:

$$\textit{Wall and glass transmission loss} = (0.8U_w + 0.20U_g)A_w \quad . \ . \ . \ . \ . \ . \quad (1a)$$

The ceiling transmission loss per degree temperature difference is equal to the ceiling coefficient (U_c) times the ceiling area, or

$$\textit{Ceiling transmission loss} = U_cA_c \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (2)$$

The ceiling coefficient (U_c) should also include the value of the roof whether flat or pitched. Consequently, the combined ceiling and roof coefficient should be used for the value of U_c.

Floor Transmission Loss

According to data in the A.S.H.V.E. Guide 1943, the floor loss for houses with basements may be neglected. Similarly, the floor loss into the ground for houses with the floor located directly on the ground is comparatively small and may be neglected. If, however, the house has no basement and is placed on piers so that the space under the floor is exposed to air at or near the outside temperature, then an allowance should be made for the heat loss through the floor into this space. The floor transmission loss per one degree temperature difference in the latter case is as follows:

$$\textit{Floor transmission loss} = U_fA_f \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (3)$$

Infiltration Losses

The air change method was used for estimating infiltration, the allowance being one air change per hour for houses without weatherstrips or storm windows and one-half air change per hour for houses with either weatherstrips or storm windows. Using this assumption, it will be found that the total heat loss will not vary appreciably in the average case from that obtained by the crack method given in the Guide, using double-hung wood windows of average width of crack. Based on an 8 ft ceiling height (assumed in this case), the heated volume of a one-story house will be $8A_c$ and the infiltration allowance for a one-story house without storm windows or weatherstrips (assuming one air change per hour) will be:

$$\left.\begin{array}{l}\textit{Infiltration loss (one-story house)}\\ \textit{(No storm windows or weatherstrips)}\end{array}\right\} = \frac{8A_c}{55.5} = 0.144A_c \qquad (4a)$$

With two-story houses, the volume will of course be greater for the same perimeter or top floor ceiling area. For example, in the case of a two-story house, the volume will be equal to the flat or projected top floor ceiling area (A_c) times the sum of the ceiling heights of the two floors, or for an 8 ft per floor ceiling height, the volume will be $16A_c$ and the infiltration allowance for a two-story house without storm windows or weatherstrips (assuming one air change per hour) will be:

$$\left.\begin{array}{l}\textit{Infiltration loss (two-story house)}\\ \textit{(No storm windows or weatherstrips)}\end{array}\right\} = \frac{16A_c}{55.5} = 0.288A_c \qquad (4b)$$

Since for houses with storm windows or weatherstrips, the infiltration allowance is one-half an air change per hour instead of one air change per hour, the infiltration losses with storm windows or weatherstrips will be one-half the losses for houses without storm sash or weatherstrips given in equations 4a and 4b, or $0.072A_c$ and $0.144A_c$ respectively for one- and two-story houses.

An 8 ft ceiling height was assumed for estimating the infiltration allowance because this ceiling height is becoming more or less standard practice in current low-cost housing. However, ceiling heights of 8½ and 9 ft were commonly used in the older houses, but it will be found that the glass area percentage in these older houses generally averages less than 20 per cent, which compensates for the higher ceiling heights. Even disregarding any possible compensating effect due to a less-than-20-per-cent glass area, an actual ceiling height of 9 ft instead of 8 ft, as assumed for the chart, would increase the calculated infiltration allowance by ⅛ or 12½ per cent. Inasmuch as the infiltration allowance may be only a small percentage—10 to 20 per cent in many cases—of the total heat loss, the error due to the assumption of an 8 ft ceiling height will usually be comparatively small when considered from the standpoint of the total heat loss of the building.

Total Heat Loss

The total heat loss per degree temperature difference is of course equal to the summation of the wall and glass transmission loss, the ceiling (plus roof) transmission loss, the floor transmission loss (if any) and the infiltration loss. For example, the heat loss per degree temperature difference (H_d) for a one-story house without storm windows or weatherstrips and with a basement (neglecting the floor loss) using the values in equations (1a), (2) and (4a), but omitting (3), will be:

$$H_d = (0.80U_w + 0.20U)A_w + U_cA_c + 0.144A_c = (0.80U_w + 0.20U_g)A_w + (U_c + 0.144)A_c \qquad (5)$$

This formula can be further simplified by substituting the value of $U_g = 1.13$ (for single glass), in which case the quantity $0.20U_g$ becomes 0.226 and the formula for a one-story house with basement and without storm windows and weatherstrips then becomes:

$$H_d = (0.80U_w + 0.266)A_w + (U_c + 0.144)A_c \qquad (5a)$$

In a similar manner, equations may be developed for one-story houses with

storm windows or weatherstrips and for two-story houses and also for houses without basements. Wherever the floor heat loss is to be included, the quantity ($U_f A_f$) given in equation (3) will appear. For example, the equation for a one-story house similar to that for which equation (5a) is applicable but without basement and with the floor above ground so that there is an air space underneath, will be as follows:

$$H_d = (0.80 U_w + 0.226) A_w + (U_c + 0.144) A_c + U_f A_f \quad \ldots \ldots \quad (6)$$

HEAT LOSS CHARTS

The chart (Fig, 1) was developed by plotting these and other similar equations. For instance, Scale A for ceilings is a plot of the quantity (U_c +

TABLE 1—SCALES OF FIG. 1 TO BE USED FOR CALCULATING HEAT LOSSES

Stories	Basement	No Weatherstrips			With Weatherstrips		
		Ceiling	Wall	Floor	Ceiling	Wall	Floor
No Storm Windows or Doors							
One-Story Houses	With basement[a]	A	D		B	D	
	No basement[b]	A	D	F	B	D	F
1½-Story Houses	With basement[a]	C	D		A	D	
	No basement[b]	C	D	F	A	D	F
Two-Story Houses	With basement[a]	C	D		A	D	
	No basement[b]	C	D	F	A	D	F
With Storm Windows and Doors							
One-Story Houses	With basement[a]	B	E		B	E	
	No basement[b]	B	E	F	B	E	F
1½-Story Houses	With basement[a]	A	E		A	E	
	No basement[b]	A	E	F	A	E	F
Two-Story Houses	With basement[a]	A	E		A	E	
	No basement[b]	A	E	F	A	E	F

[a] Also for houses without basement but with first floor directly on ground.
[b] Floor above ground with air space between.

0.144)A_c, which appears in equations (5), (5a), and (6) and is applicable to one-story houses without storm windows or weatherstrips. This quantity of course includes the infiltration loss. The other scales in Fig. 1 are plotted from the following equations:

Scale B (for ceilings): $(U_c + 0.072) A_c$.
Scale C (for ceilings): $(U_c + 0.288) A_c$.
Scale D (for walls): $(0.80 U_w + 0.226) A_w$ (for single glass).
Scale E (for walls): $(0.80 U_w + 0.09) A_w$ (for double glass).
Scale *F* (for floors): $U_f A_f$.

How to Use Chart

The solution of any heat loss problem by means of this chart involves first the selection of the proper scales from Table 1. These of course depend upon the requirements of the problem.

The next step is to locate the ceiling (plus roof) coefficient on Scales A, B, or C and draw a line to point X, calling this Line 1. Next find the ceiling area on the left hand scale and draw a horizontal line to Line 1, and then a line vertically downward to the horizontal scale. The result indicated will be the ceiling heat loss per 1 deg temperature difference. This result includes the infiltration allowance.

The wall and glass heat loss is determined in a similar manner using Scales D or E. If the floor heat loss is to be included (as indicated by Table 1), this may be determined by means of Scale F and the left hand area scale, as with the ceiling heat loss.

The wall and ceiling—or wall, ceiling, and floor—losses, as the case may be, should be added, the result being given in terms of the heat loss for the building per 1 deg temperature difference. This should be multiplied by the total temperature difference for the location of the building. The gross exterior wall area (based on inside dimensions) and including doors and windows should be used in connection with Fig. 1. Similarly, the ceiling area is based on inside dimensions.

Example 1: Calculate the heat loss of a 20 x 30 ft (inside dimensions) two-story uninsulated frame residence with basement and without storm windows or weatherstrips located in Chicago. The ceiling height is 8 ft. The wall coefficient is assumed to be 0.25 and the ceiling-roof coefficient is assumed to be 0.32.

Solution: According to Table 1, Scales C and D of Fig. 1 should be used for this problem. Scale F is not used because the residence has a basement and the floor heat loss is therefore neglected. The solution of this problem is indicated by the dotted lines on Fig. 1. The gross wall area is 1600 sq ft and the ceiling area is 600 sq ft. It will be noted that the ceiling heat loss is approximately 367 Btu per hr per deg temperature difference. This result includes the infiltration allowance. The wall and glass heat loss, according to Fig. 1, is 678 Btu per hr per 1 deg temperature difference. The sum of these quantities is 1045 Btu per hr per 1 deg temperature difference. Assuming an inside temperature of 70 F and an outside temperature of −10 F, the temperature difference will be 80 F and the design heat loss will be 80 x 1045, or 83,600 Btu per hr.

Although this chart is intended specifically for residences, it can be adapted in certain cases to other types of structures such as small office buildings or factories. Similar charts may however be developed for other structures.

Example 2: Assume a small one-story factory, 20 x 50 ft, having a ceiling height of 8 ft, plain 8 in. brick walls and a 4 in. concrete roof deck with 1 in. of insulation, located in New York and exposed on all sides.

Solution: The coefficients are respectively 0.50 for the walls and 0.18 for the roof. The wall area is 1120 sq ft, the ceiling area is 1000 sq ft, and the heat losses according to Fig. 1 using Scales A and D will be 320 (ceiling) plus 695 (walls) or 1015 Btu per hr per deg temperature difference. If the tem-

perature difference is 70 deg, the design heat loss will be 1015 x 70, or 71,050, or say 71,000 Btu per hr.

ESTIMATING ANNUAL FUEL CONSUMPTION

The approximate average annual fuel consumption can be estimated by multiplying the heat loss per 1 deg temperature difference (H_d) obtained from Fig. 1, by the number of thousand degree-days and by the factors given in Table 2 for various fuels. The assumed efficiencies and calorific values on which these factors are based are indicated in the table so that the factors may readily be modified on the basis of other efficiencies and calorific values, if desired. While the efficiencies in this table agree substantially with the values in use by a local public utility, it is realized that there may be a wide variation in combustion efficiencies encountered in actual practice and where specific data are available, the factors in Table 2 should be modified accordingly.

TABLE 2—FACTORS FOR ESTIMATING AMOUNT OF FUEL REQUIRED PER AVERAGE HEATING SEASON, BASED ON INSIDE TEMPERATURE OF 70 F

These factors when multiplied by the heat loss of the building per 1 deg temperature difference (H_d) and by the thousands of degree-days per heating season will give the approximate amount of fuel required per average heating season based on an inside temperature of 70 F.[a]

FUEL	ASSUMED CALORIFIC VALUE	METHOD OF FIRING	ASSUMED HEATING EFFICIENCY	FACTOR	RESULT WILL BE
Coal	13,000 Btu	Hand-fired (Residence)...	0.50	0.00185	Tons of Coa
		Stoker (Residence).......	0.60	0.00154	
		Stoker (Industrial).......	0.70	0.00132	
Oil	141,000 Btu per gal	Conversion..............	0.60	0.284	Gallons of Oil
		Oil-Designed............	0.75	0.226	
Manufactured Gas	535 Btu per cu ft	Conversion.............	0.65	0.069	Thousands Cubic Feet Manufactured Gas
		Gas-Designed...........	0.80	0.056	
Natural Gas	1000 Btu per cu ft	Conversion..............	0.65	0.0369	Thousands Cubic Feet Natural Gas
		Gas-Designed...........	0.80	0.030	

[a] Example: The heat loss of a certain building (from Fig. 1) is 1200 Btu per hr per degree temperature difference and is located in a city having 6500 degree-days. The probable fuel consumption in coal (hand fired) will be $0.00185 \times 1200 \times 6.5 = 14.4$ tons.

Degree-days calculated on a 65 F base are applicable to an inside temperature of 70 F. Consequently the factors in Table 2 are applicable only to this inside temperature. For lower inside temperatures, the fuel consumption will be correspondingly less, and vice versa. For other *average* inside temperatures than 70 F, the estimated fuel consumption can be adjusted on the basis of the ratio of the inside-outside temperature difference. For example, if the average

day and night temperature is 65 F (instead of 70 F) and the building is located in a city where the average outside temperature during the heating season is 30 F, the fuel consumption will be reduced in the ratio of (70-65)/(70-30) or 5/40 or 12½ per cent. Average outside temperatures during the heating season (assumed to be from October 1 to May 1, or seven months) are given in Table 2 on pp. 132 and 133 of the Guide 1943. There will be a slight error in some cases when using these average outside temperatures because of the fact that the actual heating season may be of greater or lesser duration than seven months.

LIMITATIONS OF CHART

As previously stated, this chart is intended for one, and one one-half, and two-story houses, but it may also be used for small one and two-story office buildings and factories, if the conditions are similar to those on which the chart is based. It is not expected, however, that every one or two-story residence heat loss problem can be solved by this type of chart as special or unusual conditions will be encountered.

A two-story house with a heated attic would be considered to be a three-story house so far as the application of the chart is concerned. Since a maximum of two stories is provided for by the chart, it would be necessary to add another scale to the chart for three-story houses, although the error would be small if the problem were solved as if the house were of two stories, provided the proper gross wall and top floor ceiling areas are used. The actual error resulting from this assumption would be that due to the infiltration allowance being for a two-story house instead of for a three-story building.

Unheated garages and sleeping porches should be neglected and the building considered as if they were not a part of the structure. Heated garages and sleeping porches maintained at room temperature should of course be figured the same as other heated rooms or spaces. If the temperature maintained in such spaces is lower than the inside room temperature, the heat losses from such rooms or spaces will be correspondingly reduced. If these spaces are comparatively small and are to be heated at all, it will usually be sufficiently accurate to consider them to be maintained at room temperature.

Since no allowance is made for heated basements used, for example, for living quarters, the heat losses from such spaces may be considered by adding the basement floor and wall heat losses to the heat loss for the remainder of the structure, basing the calculations on the proper surface areas, coefficients, and temperature differences.

DISCUSSION

H. M. BETTS, Minneapolis, Minn. (WRITTEN): This paper provides a long needed short cut method of determining heat losses of smaller structures with a reasonable degree of accuracy. Contractors in submitting bids for heating installations have been forced to calculate heat losses the hard way or to run the risk of guessing wrong and being compelled to complete a contract at a loss. The time spent on heat loss calculations is, of course, lost if they fail to obtain the contract. Many have developed short cut methods of their own but these are based mostly on other

buildings of comparable size or some rule of thumb which may be anything but accurate.

As the author states, the method which he has developed should be used with some discretion and should not supplant the regular method as outlined in The Guide. It will, however, find a very definite place in the heating trade and with public utilities. Since it is based on formulae and information contained in The Guide, it will provide a standard time-saving method of determining approximate heat losses in the same way that the method outlined is The Guide is looked upon as the standard method today. The author is to be commended for his work.

G. A. ERICKSON,[1] St. Paul, Minn. (WRITTEN): The graphical method presented in this paper for calculating heat losses for small buildings is by far the simplest method we have seen devised.

In roughly checking the author's method with the standard method of determining heat losses used in The Guide, we found the hourly Btu loss results in reasonable agreement and sufficiently accurate for many purposes. The percentage error in some cases was, however, slightly more than the plus or minus 3 per cent mentioned in this paper. This variation between the two methods (which incidentally was larger for insulated houses than uninsulated houses) was due perhaps to the fact that we took into account the floor loss to the basement and used the crack method to determine infiltration loss. We found the results of the two methods approximately the same, however, when we based the infiltration loss on ½ or 1 air change per hour and when we disregarded the floor heat loss in those houses having basements.

We are in agreement with the use of the combined roof and ceiling coefficient for determining heat loss of this portion of the building because of facts brought out in a previous report.[2] A study of the actual air temperatures in the ventilated attics of these four test houses, as compared with the calculated attic air temperatures, indicated the importance of considering the insulation value of the roof. The close correlation obtained between the actual and calculated heat losses of these test houses was due in part to the consideration of heat losses through the combined ceiling and roof, and also to the use of proper basement temperatures in figuring the floor heat loss. The actual basement temperatures recorded in the test houses showed the average unheated basement air temperatures to be 53 F to 57 F during below zero weather. We, therefore, believe the author's method should make some allowance for floor heat loss for houses with basements, especially in colder climates. If the floor loss is neglected, the percentage of error of the final results between the author's method and the standard method (floor loss considered) will, however, be smaller for houses in warmer climates and for houses of 1½ or 2 stories rather than 1 story.

The problem of figuring heat loss of 1½-story houses with dormers and shed roof is naturally more complicated than 1 or 2-story buildings because of the broken up areas and variation in wall, ceiling and roof construction. We assume the flat and sloping ceiling areas are to be grouped together and an average combined roof and ceiling coefficient used. On this basis we checked the 1½-story test house (using the author's charts) and found the results fairly accurate when the infiltration loss of the standard method was figured as ½ air change per hour. When the crack method was used for infiltration loss, the variation was increased to slightly more than 3 per cent.

The use of the graphical method for determining heat loss calculation as given in this paper should prove valuable to heating and ventilating engineers, architects, contractors, and others, and is to be especially recommended as a rapid method of checking heat loss, and heating and insulation requirements of residences and other small buildings.

[1] Wood Conversion Co., St. Paul, Minn.

[2] Heat Loss Studies in Four Identical Buildings to Determine the Effect of Insulation, by D. B. Anderson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942.)

H. M. HART, Chicago, Ill. (WRITTEN): This report may serve as a means of making a rough calculation of fuel requirements for one- or two-story dwellings having 8 ft ceilings and a combined window and outside door area equal to 20 per cent of the outside wall area.

However, I feel that it would be a step backward for this Society to indorse any method based on so many assumptions. For many years we have been concentrating on trying to raise the level of heating engineering by removing the guesswork, and I think a lot of progress has been made.

My early childhood was spent in a house heated by steam radiators. The radiators were constructed of 1 in. pipe screwed into cast-iron bases. The sizes were selected on the basis of 1 tube 30 in. long to 60 cu ft of space to be heated.

My recollection is that we were always warm enough, but I also remember the woolen underwear. The house is still occupied by members of my family and the old radiators are still doing the heating, although the second-floor ceiling has been insulated and storm windows and storm doors are now used. I know it was the best heating system in the town in 1891, but I also know that this is not true today.

For a rough calculation of fuel requirements the Fuel Rationing Division of the OPA has certainly adopted a simple formula, and I am sure many of us have found how rough it could be.

The method suggested by the author is certainly an improvement over that, but I would still hesitate to suggest that it be adopted by OPA for fuel rationing. The idea is good, but it is too limited in its application, and I fear that it might be misused.

The *Heating, Piping and Air Conditioning Contractors National Association* have found charts preferable to graphs, because of greater simplicity and possibility of longer use before replacement. All calculations were on the chart and they were based on Guide requirements.

E. VERNON HILL, Chicago, Ill. (WRITTEN): I believe this paper will find wide use, and the author is to be commended for this ingenious method. I have not had the time so far to study the paper as it deserves, but I gather the impression from reading it over that the curves A, B, C, D, E, and F are plotted from simplified heat transmission formulae. Ordinate and abscissa on the chart represent wall and floor, roof areas, etc., and heat losses. Drawing a horizontal line from the area scale to a diagonal line from the appropriate factor on the curve to zero and dropping down from this intersection to the heat loss scale is the total heat loss for the area under consideration. Figuring heat losses, therefore, by the author's graphic method is as simple as that.

Of course, the danger in using this as well as other simplified methods of making calculations is that the one who uses it is liable to forget the proper use of fundamental methods of calculation and the formulae as set forth in The Guide; but even so, the saving in time and grey matter is still, I think, amply justified. Most engineers and heating contractors are willing to leave the extremely accurate and technical calculations to our college professors and research men; and if the results will check within plus or minus 3 per cent with the older methods of calculation, I feel that this is close enough for all practical purposes. When we consider the coefficients of transmission as they are in The Guide today and compare them with the same coefficient or the coefficients for the same material of 10 years ago, we can excuse any slight discrepancy in values as determined by the chart prepared by the author.

One thing that interests me in examining Fig. 1, which is the full page chart under discussion, is the following: The scales for ceiling, floor, wall areas, etc., and the scales for heating losses are identical on both charts. It would appear desirable, therefore, to combine the two in one chart, thus simplifying the entire procedure. I note in superimposing the two, the curves A and E come very close together but

could not, I believe, be a serious obstacle to combining the two charts. Possibly there are other reasons I do not appreciate that the author will explain.

JOHN JAMES, Cleveland, Ohio (WRITTEN): An ingenious chart has been presented for quickly calculating the heat loss of a residence. This graphical method should serve as a very useful tool in four types of problems such as, (1) estimating the yearly fuel consumption of a structure, (2) determining heat loss savings resulting from the application of insulation and storm windows, (3) sizing and applying conversion automatic heating equipment to existing structures, and (4) making rough estimates of the total cost of a heating or air conditioning system.

Short-cut formulae and charts such as the one presented in this paper have been advocated for the purpose of reducing the engineering charges on an installation. Of course, it is readily apparent that this chart is not useful in designing a heating system, as it is necessary to calculate the heat loss of each individual room for properly selecting the required amount of radiation or air supply.

Many have expressed the feeling that The Guide method of calculating heat loss is too tedious and cumbersome. However, it can readily be shown that such an empirical formula as Carpenter's requires almost as much time to use as The Guide method. In both methods of computation, one of the most time-consuming factors is the time needed to actually measure or scale from drawings the dimensions of the room and window and door areas. After this job is completed, only a short time is required to calculate the heat loss.

It might be appropriate at this time to review briefly the empirical formula suggested by Prof. R. C. Carpenter to illustrate how closely The Guide method compares. Professor Carpenter's formula is as follows:

$$H = (1\,G + 0.25\,W + 0.02\,nC)\,(t_0 - t_1)$$

Practically the only difference between this formula, which incidentally many engineers are now advocating in some such form, and The Guide method is that more exacting values of U (overall heat transmission coefficient) are substituted such as 1.13 instead of 1 for single glass, and the actual U value for the wall under consideration instead of the figure 0.25 which is for an ordinary wall. True, this formula does not make use of the crack method for estimating infiltration, but it will be recalled that The Guide suggests that the amount of air leakage can be roughly estimated by assuming a certain number of air changes which can be substituted for n in Professor Carpenter's formula.

Summarizing, it is my opinion that the graphical method presented in this paper will serve as a very useful supplement to The Guide procedure for the purposes previously outlined. Until some method can be devised which will eliminate the necessity of actually measuring a room to determine the heat loss, it is believed that THE GUIDE method offers distinct advantages over any empirical formula now suggested.

R. K. THULMAN, Washington, D. C.: The results obtained by the author's graphical method were in fairly close agreement with short cut methods developed by the FHA. In developing a formula for heat loss the FHA had found a reasonably consistent relationship between all the area through which heat is lost, the cubical contents, and the floor area and therefore, especially since floor area is used as a basis of other computations having to do with the property valuation and underwriting, FHA had adopted floor area as the single factor for determining heat loss. An analysis of 36,000 cases had shown the FHA floor area formula to be consistent with accurate methods for determining heat loss.

The formula method, in my opinion, can be more simply applied, and avoids the additional computation of gross wall area necessary in the author's method. I also find that a formula or table is more readily understood by architects than the author's graphical device.

H. E. LEWIS, Toledo, Ohio: I think the chart is very helpful and there is one incidental point that I would like to make. The author has offered some very

helpful operating efficiencies for heating systems. Operating efficiencies are one major variable which are often overlooked in working out heat loss calculations. The variation of from 50 per cent to 60 per cent in the efficiency of the heating system makes a difference of about 20 per cent in calculated heat losses, whereas other factors may affect the loss only about 3 to 5 per cent. This is a worthy contribution to bringing about uniformity in assumed heating efficiencies for coal, oil and gas, for different firing conditions.

C. M. HUMPHREYS, Pittsburgh, Pa.: The author has made a very ingenious chart for calculation of heat losses, but I think we should recognize very definitely its limitations. All the discussions on it have dwelt around the fact that it is for residences, and I think that the author definitely states that that is what it is for although there is nothing on the chart to so indicate. In deparing from the conventional type of building wide variations in the relationship between glass, wall, and so on, would be obtained and therefore serious errors may be introduced. Some time ago the author sent me the chart and I immediately tried the method out on the office with a resulting 18 per cent error, based on method of calculation used in THE GUIDE. That was due to the fact that the ratio of glass to wall in that particular room was quite different from that of a normal residence.

It should also be pointed out a little more clearly that this method of calculation is not applicable to individual rooms in a house but only to the structure as a whole. A chart of this kind should very definitely indicate upon it its limitations, otherwise it is so likely to be used for a purpose for which it was never intended; that will almost certainly result in trouble.

It has been stated that the method of heat loss calculations recommended by THE GUIDE is lengthy. A well-planned form for heat loss calculations can do much to reduce the time required, and I would suggest the development of such a form as a first step to shorter and accurate heat loss calculations.

AUTHOR'S CLOSURE: Mr. Hart referred to the assumption of a 20 per cent glass and door area and the 8 ft ceiling height, intimating that the chart was applicable roughly only for these conditions. The approximate variations in the total heat loss per one per cent deviation in percentage of glass and door area from the assumed 20 per cent, as for example 21 per cent instead of 20 per cent, for 20 x 30 (inside dimensions) one and two-story residences with and without storm windows are as follows:

STORIES	COEFFICIENTS	NO STORM WINDOWS PER CENT	WITH STORM WINDOWS PER CENT
One Story	$U_w = 0.1$ $U_c = 0.32$	1.5	0.16
	$U_w = 0.25$ $U_c = 0.32$	1.2	0.28
Two Story	$U_w = 0.25$ $U_c = 0.32$	1.5	

It is apparent therefore that there will be comparatively small variations in the total result due to small deviations from the 20 per cent average assumed, especially in the case of houses with storm windows. As pointed out in the paper, the combined glass and door areas of most residences range between 15 per cent and 25 per cent.

As to the effect of variations from the 8 ft ceiling height, this would affect only the infiltration loss and not the transmission loss since the actual gross wall area

based on the actual ceiling height is used in arriving at the transmission loss. An actual ceiling height of 9 ft (instead of 8 ft) would mean that the infiltration loss (based on the air change method used for the chart) would be increased 1/8, but since the infiltration loss may be only 10 to 20 per cent of the total heat loss, the error would amount to only 1/8 of this 10 to 20 per cent, or say about 1 per cent to 2½ per cent of the total heat loss. However, as pointed out in the paper, the higher ceiling heights are generally found in the older structures which have small glass and door area ratios, so the two are more or less compensating. Ceiling heights of 8 feet are being generally used today and the glass and door areas of present day construction will average closer to 20 per cent.

The important point is that heat losses cannot in my opinion be calculated any more accurately than the result obtained by this chart, regardless of the method used, primarily because of the difficulty of estimating the infiltration loss accurately, as I shall mention later.

Dr. Hill asked whether these two charts could not be superimposed on one graph. That could be done. They were separated for convenience, and, if possible, to avoid some confusion.

Mr. Erickson refers to basement losses. The basement losses are neglected in this chart for residences where the basement is not heated. It was found that the loss through basement floors is very small. In fact, the coefficient is only 0.10 per degree temperature difference, or 2 Btu for the total temperature difference between the air and the floor, and the heat loss from the first floor rooms into the basement is correspondingly very small, and can be neglected. If you take an actual case and calculate the actual basement losses you will find that they are a very small part of the total if the basement is below grade.

I want to say a word here on the accuracy of the air change *vs.* the crack method. It is generally considered that the crack method of figuring infiltration is more accurate than the air change method, by which an arbitrary allowance is made for infiltration, based, of course, on the volume of the space under consideration. In years past I have been a staunch proponent of the crack method of figuring infiltration, but I have changed my ideas somewhat on this subject, because I doubt if it is possible to estimate air leakage accurately, regardless of the method used. In Chapter 5 of the HEATING, VENTILATING, AIR CONDITIONING GUIDE, factors are given for infiltration through windows, which are based on a certain wind velocity through various widths of cracks and types of windows. For example, the infiltration for a double hung window with a 1/16 in. crack and 3/64 in. clearance is 21.4 cu ft of air per foot of crack for a 10-mile wind velocity.

Now, I doubt if anyone knows what the width of a crack is for a building, new or old, before or after it is built. Not only that, but, according to this table, if they should be a little off in the prognastication as to what the width of crack is, they may be way off in the amount of infiltration allowance.

For example, if the crack is 3/32 of an inch, instead of 1/16 of an inch, the infiltration for a 10-mile wind velocity will be 68 cu ft, or more than three times as much. I do not think anyone can guess what this width of crack is within that degree of accuracy.

Furthermore, for maximum accuracy it is necessary to take into consideration all the other factors that enter into infiltration such as loss through open windows and chimneys and back pressure due to doors being closed, and so on. So the question of making allowances for infiltration is not subject to meticulous exactitude. You could easily be as much as 200 or 300 per cent off on the infiltration allowance.

Mr. James referred to the use of the chart for individual rooms. It can be used for individual rooms for one-story residences. It is not adapted to two-story residences.

Any similarity between this method and that adopted by the FHA, as referred to

by Mr. Thulman, is purely coincidental. I was wholly unaware and oblivious of the fact. The basis of derivation of the chart is given in the paper.

It is true that the heating efficiency assumed may vary the results considerably. If you have actual efficiencies in specific cases other than those given in the paper, use your own efficiency for estimating the amount of fuel required.

Mr. Humphreys referred to variations in the result due to variations in the percentage of glass area from the 20 per cent used in the paper which I have already covered. If you will take some actual cases you will find that there is a comparatively small difference due to variations in the percentage of glass area, provided you use the gross wall area (including doors and windows) in using the chart.

In Memoriam 1943

Name	Joined
Theodore H. L. Backus	1916
Waldo Mumford Bailey	1930
Harry V. Bayse (*Life Member*)	1923
Willard B. Graves (*Life Member*)	1906
Jeremiah J. Herlihy (*Life Member*)	1914
Benjamin H. Jessup	1937
Arnold R. Kamman	1921
James T. Machen	1934
Herman F. Maier (died October 25, 1942)	1926
Felix N. Parsons	1943
John J. Raine (*Life Member*)	1912
R. W. Rodman	1922
Henry W. Skinner	1920
William H. Stangle (died August 7, 1942)	1940
J. E. Swenson	1930
Harold Luttrell Temple (Flying Officer, R.A.F., killed in action August 10)	1942
Woodworth Wethered	1938
Peter P. Wolff	1935
A. H. Woolston	1919

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